BBC RD 1989/2

Research
Department
Report

THE CONTROL OF THE AUDIBLE EFFECTS OF GROUND VIBRATIONS IN BUILDING STRUCTURES

R. Walker, B.Sc.(Eng), A.M.I.E.E., and C.D. Mathers, M.Sc., C.Eng., M.I.E.E., M.I.O.A.

Research Department, Engineering Division
THE BRITISH BROADCASTING CORPORATION

BBC RD 1989/2

THE CONTROL OF THE AUDIBLE EFFECTS OF GROUND VIBRATIONS IN BUILDING STRUCTURES

R. Walker, B.Sc. (Eng), A.I.M.E.E. and C.D. Mathers, M.Sc., C.Eng., M.I.E.E., M.I.O.A.

Summary

Over a period of about eight years a large amount of investigation into the problems of vibration isolation, particularly of large structures, has been carried out, mostly in preparation for the development of a site (The Langham) which is subject to severe vibration interference from London Underground trains. Under the direction of an Acoustic Consultant, an extensive series of tests and measurements was carried out on a purpose-built, experimental test slab in the basement of the existing building on the Langham site. Other comparative measurements were made. The well-known limitations of some types of commercial vibration isolation mountings as a result of modal vibration responses have been demonstrated and, for the isolation of large structures, additional limitations due to the modal vibrations of the supported structures have been quantified. Additionally, and most importantly, the severely limiting effects of acoustic coupling have been measured to be much larger than had been suspected. The achievable isolation of a studio-like structure was limited to a maximum of about 25 dB at all frequencies, for a single-stage system. Some simple theoretical analyses of the actual, imperfect behaviour mechanisms of vibration isolation systems, when taken together with other measurements, have enabled predictions of the performance of proposed isolation systems to be made with reasonable accuracy. It is shown that an expensive double isolation system is likely to be required to satisfy the normal criterion for the drama studio background noise level on the Langham site. Subsequently, the development of the Langham site was abandoned but the results of the work are applicable to any studio site which is subject to ground-borne vibrations.

Issued under the Authority of

M. E. B. Moffat

Head of Research Department

Research Department, Engineering Division
BRITISH BROADCASTING CORPORATION

(PH–282)

1989

© BBC 2004. All rights reserved. Except as provided below, no part of this document may be reproduced in any material form (including photocopying or storing it in any medium by electronic means) without the prior written permission of BBC Research & Development except in accordance with the provisions of the (UK) Copyright, Designs and Patents Act 1988.

The BBC grants permission to individuals and organisations to make copies of the entire document (including this copyright notice) for their own internal use. No copies of this document may be published, distributed or made available to third parties whether by paper, electronic or other means without the BBC's prior written permission. Where necessary, third parties should be directed to the relevant page on BBC's website at http://www.bbc.co.uk/rd/pubs/ for a copy of this document.

THE CONTROL OF THE AUDIBLE EFFECTS OF GROUND VIBRATIONS IN BUILDING STRUCTURES

R. Walker, B.Sc.(Eng), A.I.M.E.E., and C.D. Mathers, M.Sc., C.Eng., M.I.E.E., M.I.O.A.

1. Introduction 1

2. Theoretical Considerations 2

2.1 The relationship between surface vibrations and the resulting airborne noise levels 2

2.2 Ideal vibration isolation systems 3

2.3 Limitations of vibration isolation systems 4

2.3.1 Non-ideal anti-vibration mountings 4
2.3.2 Structural resonance modes 6
2.3.3 Acoustic coupling 7
2.3.4 Differences between source isolation and receiver isolation 8
2.3.5 Performance envelope of practical vibration isolation systems 8

3. The Langham Test Slab and other Measurements 9

3.1 Vibration measurement — introduction and general considerations 9

3.2 The Langham Test Slab 10

3.2.1 Description 10
3.2.2 Measurements — vibration isolation 10
3.2.3 Measurements — airborne coupling 13
3.2.4 Measurements — Slab modes and 'Q'-factors 15

3.3 'Ideal mass' avm measurements 16

3.4 Surface-to-air coupling 18

3.5 Attenuation of vibration as a function of distance 19

3.6 Attenuation of vibration as a function of height within a building 20

3.7 Measurements at Maida Vale, Studio 7 22

3.8 Measurements in Studio B11 24

4. Further Developments of the Langham Test Site 25

4.1 Construction 25

4.2 Structural modes 25

4.3 Vibration isolation 26

4.4 Internal noise levels 26

4.5 Second enclosure 26

5. Application to the Langham Development 27

5.1 General 27

5.2 Source vibration levels 27

5.3 Permitted surface vibrations for a drama studio 28

5.4 Vibration isolation requirements 28

5.5 Comparison of the target with achieved isolations 28

5.6 Direct perception of vibration 29

6. Proposed Constructions. 29

7. An Alternative Proposal for the Isolation of the Drama Studios 31

8. Conclusions 32

9. References 33

10. Acknowledgements 34

Appendix 1
Outline of Computer Program for the Derivation of Network Responses by Inspection 35

Appendix 2
Component Values for the Idealised Equivalent Circuits 35

Appendix 3
Types of Mounts and Description 36

THE CONTROL OF THE AUDIBLE EFFECTS OF GROUND VIBRATIONS IN BUILDING STRUCTURES

R. Walker, B.Sc. (Eng) A.I.M.E.E. and C.D. Mathers, M.Sc., C.Eng., M.I.E.E., M.I.O.A.

1. INTRODUCTION

Vibrations in building structures are responsible for many different effects, depending on their amplitude and frequency and the type and use of the building. At the upper extreme, vibrations which are perceptible to the occupants of the building cause discomfort and distraction and may adversely affect equipment and instruments. Structural damage to the fabric of the building may also occur. The most violent examples of these kind of vibrations occur as a result of earthquakes. At lower levels of perceptibility, airborne noise is generated by the coupling between the vibrations of the surfaces of a room and the air. In effect, the air within a room forms a coupling medium between the structural vibrations of the surfaces and the vibrations of the ear-drum or microphone diaphragm which are perceived as acoustic noise.

In broadcasting studios and their control rooms the maximum permitted airborne noise levels are very low so that the performers, engineers and, ultimately, the audience of listeners are not disturbed by extraneous noise. Apart from any other considerations, such as design of the ventilation plant, this means that the vibrations of the room surfaces must be controlled to very low levels, far below those giving a tactile sensation.

Vibrations in the structure of a room within a building have many possible sources, one of which is the direct excitation of the structure by airborne sound energy in an adjacent area. This transmission path, consisting of the initial airborne part, a part within the structure and a final airborne part, is commonly known as the airborne sound insulation (or isolation), despite its having at least one part which is within the building structure. It is a well-documented (if, in practice, somewhat empirical) science and is not the subject of this Report.

Other sources excite the surfaces of a room directly by vibrational energy conducted entirely within the elements of the structure of the building. This Report is concerned with the control of these vibrations and especially with their control at or near the room which is affected by them (the receiving room). These vibrations are most commonly generated by the building services plant but they may also originate at some more remote point, possibly not even under the control of the building's users. One example of the latter category in London is the LTE 'Underground' train system.

At the time when the development of the Langham site in London was being considered it was realised that the proximity of the Underground tunnels to the site constituted a significant risk to the proposed studios, especially to the drama studios which are the most sensitive to outside interference. The history of train noise in the existing Broadcasting House studios and the apparent failures of previous attempts to isolate critical areas from the vibrations generated by the trains made it essential to carry out investigations before the building design progressed very far.

In principle, the isolation of vibration is perfectly straightforward. It is simply necessary to design the room or the machine or whatever to be supported by 'anti-vibration mountings' (avms) which can be obtained in an enormous range of types, sizes and specified performances from many suppliers. Of course, the structural problems of designing the supports, especially of large rooms, are not always trivial but they are generally quantifiable. In practice, experience has shown that such a simple approach does not work. Two fundamental theoretical causes of failure were known before this investigation started (modal behaviour of the structures and of the avms) and, in addition, there were always the possibilities of faults in the constructions of those previous attempts to isolate vibrations to account for their failure. Not surprisingly, it had never proved possible to dismantle any of those failed attempts in order to discover if there were any faults in the constructions. Although the two theoretical reasons were appreciated, there was no information about the magnitude of the effects on the scale of full-sized buildings to determine whether or not they were significant.

The ensuing investigations took several forms, beginning long before the detailed discussions concerning the Langham were started. These included experiments at Research Department (beginning in early 1981) on sample anti-vibration mountings and a theoretical study of the behaviour of one type of mounting, the construction (beginning approximately in early 1979) of a full-size drama studio on a vibration isolation system at Maida Vale (now

Studio 7) and some measurements of the vibration levels existing on the Langham site.

During the detailed discussions on the proposed design for the New Broadcasting Centre it became evident that the difficulties in meeting the background noise specifications were great and that the margins for error appeared to be negative. It was therefore essential to carry out much more detailed and accurate investigations on the site. To this end, a scaled-down 'model' of a studio was constructed in stages with very many detailed measurements being made on the structure and on the component parts, particularly on different types of anti-vibration mountings. Measurements were also made in existing buildings and studios which were close to the line of the train tunnels to assess the behaviour of full-size structures. All of this work was carried out under the direction of the Acoustic Consultant, Tim Smith Acoustics.

The conception and arrangement of the test facility were those of the Acoustic Consultant in association with Arup Acoustics Ltd., the Architectural Consultants, Foster Associates and the BBC representatives of the Project Control Group. The detailed structural design was by Ove Arup Partnership and the actual construction was by Bovis Construction Limited.

Apart from the work required by the Acoustic Consultant for the Langham Project, the opportunity was taken to carry out additional measurements which are also reported here. Finally, it should be noted that much of the methodology and interpretation reported here is that of the authors and that the views expressed are therefore not necessarily those of the Consultants.

2. THEORETICAL CONSIDERATIONS

The published literature on the subject of vibration isolation is extensive and detailed. In particular, Reference 1 is a tutorial paper which includes a large bibliography (231 entries). What follows here is neither detailed nor rigorous. It is intended to illustrate the features relevant to this area of application in order to permit the reader to follow the arguments. It is not intended to be an exact numerical evaluation of any particular system.

Nevertheless, many of these theoretical results are close enough to allow correction factors, derived from the results of practical measurements, to be applied to give accurate predictions for a limited area of application.

2.1 The relationship between surface vibrations and the resulting airborne noise levels

The final stage in the coupling of vibrations from a remote source into airborne noise in an enclosure is the degree to which the vibrations of the enclosure's internal surfaces cause pressure variations in the adjacent air volume. Provided that the structural vibrations do not themselves form a subjectively disturbing tactile sensation to the occupants they are not inherently significant but the resulting pressure variations will propagate as acoustic noise in the volume of the enclosure. It will be shown later (Section 5.6) that if the criterion for airborne noise level is satisfied then the direct perception of surface vibrations is unlikely to be a problem, except, possibly, at very low frequencies. The problem of direct coupling of vibrations to transducers, such as microphones, is not considered here. As a vibration isolation problem it is more economically solved by the use of individual transducer suspensions.

The effectiveness of the conversion of vibrations to noise is one of the three principal factors in the derivation of an adequate performance target for a vibration isolation system, the other two being the source vibration levels and the desired final background noise level. Any reduction in this effectiveness is reflected directly as a reduction in the demands placed on the primary vibration isolation system.

For a structure like a masonry wall, which is massive relative to the nearby air and which has hard and non-porous surfaces, the vibrations of the surface will be coupled to the adjacent layers of air with no loss of magnitude. For the purposes of calculation of sound pressure level it is most convenient to consider the velocity of the particle movements but the equality applies just as well to the amplitudes or the accelerations. The functions are periodic and all of the quantities are the rms values.

Two rather gross assumptions are necessary before the relationship between surface vibrations and sound pressure level can be derived. These are:

i) that large (relative to the wavelength of the sound) portions of the surfaces are moving with a common velocity in amplitude and in phase. This implies that the propagation will be in the form of plane waves, with no attenuation with distance.

and ii) all of these portions of the surfaces are contributing in random phase to the total sound energy in the enclosure.

These two assumptions are, in the limit, mutually contradictory but it is a reasonable expectation that, on a small scale the first will be true and, on a larger scale the second will be. It is also assumed that the distribution of vibration velocities over the surfaces of the enclosure is uniform. In a real studio the wall surfaces are far from being hard and non-porous but are usually covered in acoustic treatment. It will be shown later (Section 3.4) that the aggregate effect of all of these assumptions is reasonably close to the measured behaviour over most of the frequency range of interest, and that simple, experimentally derived correction factors can be applied in practice to allow for the failure of the assumptions at the extremes of the frequency range.

From Reference 2, Equation 10.73,

$$L_P = L_W + 10\log(1/S_x + 4/R) + 0.1 \text{ dB} \qquad (1)$$

where L_P is the rms sound pressure level *re* 20 μPa
L_W is the rms sound power level *re.* 10^{-12} W
S_x is the radiating surface area
and R is the room constant

For the case where all of the surfaces are radiating equally, S_x can be replaced by S, the total enclosure surface area.

The room constant, R, is a function of the enclosure size and the acoustic treatment. The usually accepted equation relating the reverberation time to the enclosure dimensions and the average acoustic absorption coefficient in studio-like enclosures as derived by Eyring is

$$T = -0.161\ V/S \ln(1-\bar{\alpha}) \qquad (2)$$

where S is the surface area of the enclosure
V is the volume of the enclosure
$\bar{\alpha}$ is the average absorption coefficient
and T is the average reverberation time

Re-arranging Equation 2 gives $\bar{\alpha}$ as a function of the reverberation time for a given enclosure. From Reference 2, Equation 10.54, the room constant, R, is given by

$$R = S\bar{\alpha}/(1-\bar{\alpha}) \qquad (3)$$

For a vibrating surface with 100% coupling to the air the acoustic power radiated is given by:

$$\text{Power} = \rho c v^2 \text{ W.m}^{-2} \qquad (4)$$

where ρc is the acoustic impedance of the plane wave (ρ is the density of and c the velocity of sound in the acoustic medium) and v is the rms surface velocity in m.s^{-1}.

The total radiated power if all surfaces are radiating incoherently is therefore:

$$L_W = S\rho c v^2 \text{ watt} \qquad (5)$$

or, in terms of surface acceleration

$$L_W = S\rho c a^2/\omega^2 \text{ watt} \qquad (6)$$

where a is the rms surface acceleration in m.s^{-2} and ω is the frequency, in radians.s^{-1}

Substituting in Equation 1 from Equation 4 gives an expression for the sound pressure level in an enclosure as a result of the perimeter surface vibrations:

$$L_P = 10\log((\rho c a^2 S/\omega^2)/10^{-12}) + 10\log(1/S + 4/R) + 0.1 \text{ dB} \qquad (7)$$

2.2 Ideal vibration isolation systems

Consider the mechanical system illustrated in Fig. 1, consisting of a mass supported over a rigid and massive plane by a spring. A dashpot represents viscous losses, that is a loss component proportional to the velocity difference between the two ends of the spring. The plane has a vertical sinusoidal vibration of rms velocity v_1 and the resulting rms velocity of the mass is v_2. This model underlies the claimed performance of commercial vibration isolators. The isolation, T, is defined as $20\log_{10}(v_1/v_2)$. (It may be noted that the same system could represent a vibrating machine supported on anti-vibration mountings to reduce the coupling to the structure. The same isolation equation would represent that case also, with forces replacing velocities and the 'input' and 'output' interchanged.)

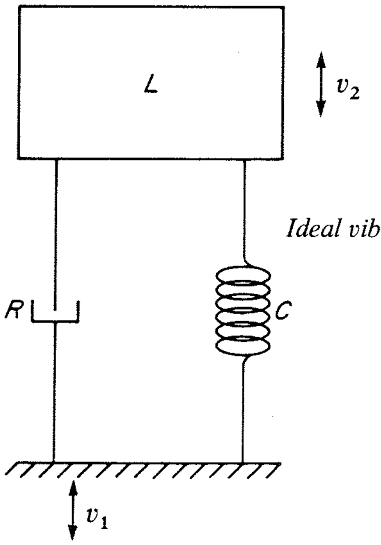

Fig. 1
Ideal vibration isolation system.

Although the solution for the response of the model in Fig. 1 can be obtained from the differential equations representing the mechanical components it is far easier, especially for electrical engineers, to transform the system to an electrical equivalent circuit. The wealth of familiar techniques developed for circuit analysis can then be applied directly. Many equivalents are possible, the most common being the impedance analogy which equates electrical voltage to force and electrical current to velocity. In this convention mass is represented as inductance, L, and compliance by capacitance, C. Resistance, R, represents an energy loss mechanism.

The simple mechanical circuit of Fig. 1 can be redrawn as the equivalent electrical circuit of Fig. 2. The response of this network can be derived by inspection if the damping component is relatively small. Fig. 3 shows the resulting isolation characteristic. At low frequencies the circuit has zero isolation ($v_1 = v_2$). At a frequency of $f_0 (= 1/2\pi(LC)^{1/2})$ the response will have a minimum with a value governed by the 'Q-factor', Q, of the network and an isolation of less than zero. Not until a frequency of f_1, which is approximately $1.4 f_0$, does the isolation return to zero. At higher frequencies the isolation increases at a rate of 12 dB/octave until a frequency of $f_2 (\approx Q.f_0)$ when the slope of the response changes to 6 dB/octave as a result of the damping component.

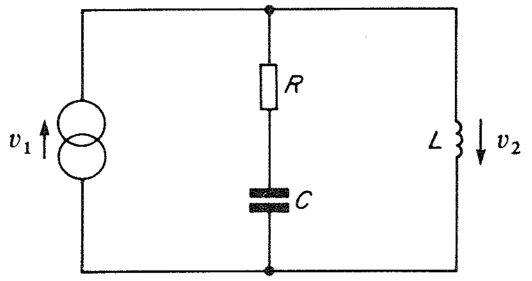

Fig. 2 – Electrical impedance analogy of the ideal vibration isolation system.

This general response is typical of the theoretical responses of all simple isolation systems, provided that the damping coefficient is reasonably small. Thus, a simple vibration isolation system is characterised, at least in theory, by a resonant frequency at and below which the isolation is negative or zero. Only at frequencies above about 1.4 times the resonant frequency is useful, positive isolation achieved. Even then, the isolation increases with frequency from zero, initially at a rate of 12 and then at 6 dB/octave. Thus, to achieve large values of isolation at any frequency requires the fundamental resonance to be located at a much lower frequency.

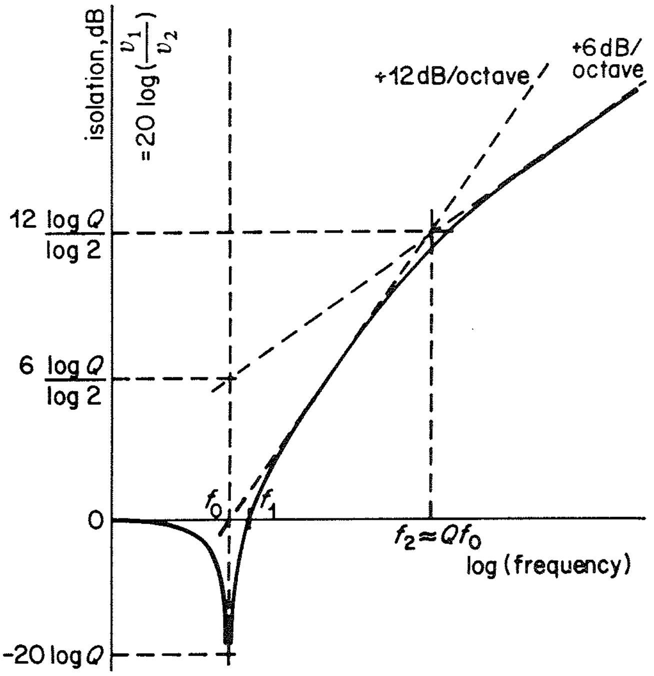

Fig. 3 – Theoretical response of the ideal vibration isolation system.

More complicated isolation systems, perhaps comprising several stages, can be analysed theoretically using the same principles. The equivalent circuits will then very likely be too complex for the simple analysis by inspection described above. The formal methods of circuit analysis and, perhaps, filter design might then be appropriate. However, it will be seen later (Section 2.3) that computer-based numerical methods are also effective at producing results based on circuit inspection. Appendix 1 gives a general description of a program, running on a small desk microcomputer, which can use inputs derived by inspection from an equivalent circuit to calculate the frequency response of the network. Running in BASIC it can plot the response of circuits with about 20–30 elements at the rate of about two frequencies per second.

2.3 Limitations of vibration isolation systems

2.3.1 Non-ideal anti-vibration mountings

The simple theory of Section 2.2 assumed that the anti-vibration mountings can be accurately represented by a single compliance with some viscous damping. In practice, the construction of the mounting inevitably involves finite dimensions and mass, in addition to its principal property of compliance. The combination means that the mounting can itself be considered as a transmission line rather than as a single, pure component. If it is lossless then the transfer impedance of the equivalent transmission line can take all values of reactive impedance from $+j\infty$ to $-j\infty$ over a range of frequencies. Alternatively, the mounting itself may consist of a series of masses and compliances; rubber-based isolators are often of this form with metal sheets within the body of the rubber. The object of the metal sheets is to obtain the appropriate degrees of stiffness and load-bearing but the effect is to add several distinct resonances to the mount impedance. Helical metal springs come between these two extremes with a very large but probably not infinite number of resonance modes.

Obviously, these types of impedance do not give the same vibration isolation as a pure compliance. As an indication of the theoretical effects of these modes consider the circuits shown in Fig. 4. The simple ideal of Figs. 1 and 2 has been modified by dividing the compliance of the mounting into three equal parts with two intermediate steel plates of mass L_1. This is representative of a common type of rubber mounting installed, for example, as roadway bridge supports[3]. They have also been used as anti-vibration mountings for entire buildings[4,5,6,7]. Fig. 5(a) shows the isolation response with values representative of the supporting of a typical drama studio. The numerical values used for the components and their derivations are given in Appendix 2.

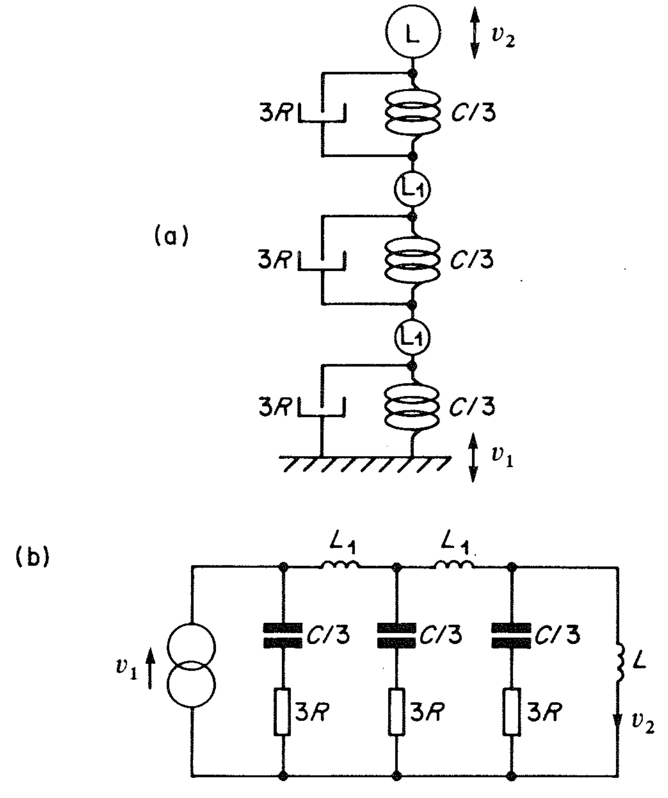

Fig. 4 – Vibration isolation system with resonance modes in the mounting.
(a) Mechanical circuit.
(b) Electrical equivalent circuit.

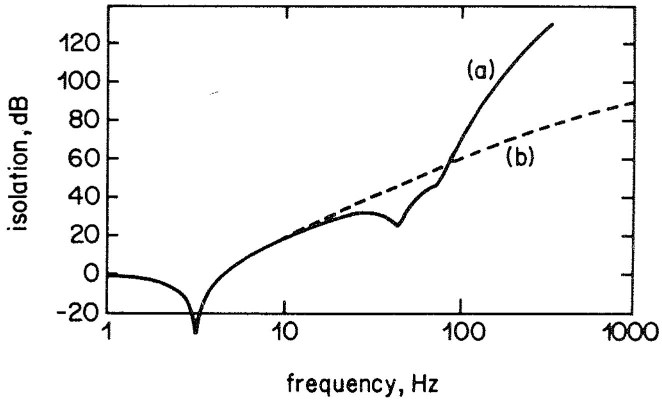

Fig. 5 – Response of vibration isolation system with resonance modes in the mounting.
(a) Response of the circuit of Fig. 4.
(b) Response of the ideal system, for comparison.

The response was obtained using the computer programme described in Appendix 1. Also shown, in Fig. 5(b), is the response which would have been obtained if the mountings had had no resonance modes. The effect of the modes is clearly visible. In this example the unwanted modes occur at much higher frequencies than the fundamental resonance, in a frequency region where there may be no requirement for isolation, but similar modes have been observed at frequencies of 50 Hz in metal helical springs of the dimensions necessary for the supporting of entire studios. Section 3.3 describes some measurements of these impedances and gives the results obtained for several different types of mountings.

The problem is recognised by the suppliers of vibration isolation components, especially the suppliers of helical metal springs, although they do not acknowledge the problem in their sales catalogues. Their solution is the use of so-called 'noise-stop' pads. These consist of supplementary compliances, usually in the form of thin rubber or glass-fibre pads mechanically in series with the main mounting. The supposed principle of operation of these pads is that they introduce an additional compliance and, therefore, together with the mass of the mounting itself, an additional resonance frequency. This will isolate the energy at the high frequencies where the main mounting fails. They also, supposedly, introduce damping to limit the severity of the main-mounting resonances because they are claimed to be made of relatively high-loss compounds. Consideration of the equivalent circuit for such an arrangement shows that these are unlikely to be significant factors. Adding the 'noise-stop' pad and the mass of the mount to the simple circuits of Figs. 1 and 2, the circuits of Fig. 6 are obtained. L_1, C_1 and R_1 represent the added components of the mass of the mount and the compliance and damping of the 'noise-stop' pad respectively.

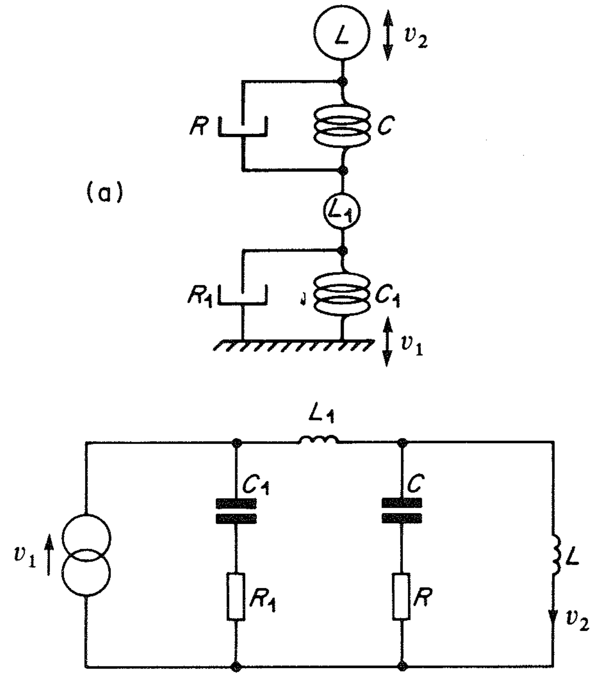

Fig. 6 – Ideal vibration isolation system with a 'noise-stop pad'.
(a) Mechanical circuit.
(b) Electrical equivalent circuit.

Inevitably, the mass of the mount, L_1, is much less than that of the supported mass, L, and the compliance of the 'stop' pad, C_1, is much less than that of the main compliance, C. Inspection of Fig. 6(b), the electrical equivalent circuit, shows that, with these two conditions, the additional components have very little effect on the overall performance except to alter the characteristic frequencies by a small amount. In particular, no amount of extra damping at the point represented by R_1 will significantly affect the overall damping of the circuit because it is (electrically) in series with a high impedance. In practice, the materials of which these pads are usually made are not of a particularly high damping coefficient anyway.

Philosophically, this can be thought of more simply. If the main mount can take all reactive impedance values from $+j\infty$ to $-j\infty$ then adding another series reactive impedance will not alter these limits but it may slightly alter the frequency at which any particular total impedance occurs. As for the damping, it is improbable that the insertion of a relatively small amount of material in compression in this way could significantly affect the energy losses of a much more massive system.

The similarity between the equivalent circuits of the ideal mountings with 'noise-stop' pads, Fig. 6, and the mounting with inherent modal resonances, Fig. 4, is evident. The effects on the frequency response of the isolation are also similar; Fig. 5 can therefore stand as being representative of either, in general terms.

2.3.2 Structural resonance modes

Section 2.3.1 considered the departures from the ideal behaviour of one of the components of a simple vibration isolation system. The mass which is supported, that is, the studio or the machine, may also have non-ideal behaviour. Above some limiting frequency the supported mass will cease to behave as a rigid body, moving as a whole, and will begin to move differently at different points. The frequency at which this will begin to happen will depend on the size of the body, its structure and the material of which it is made. At such frequencies the effective load on the anti-vibration mountings will not be the mass of the body and the achieved isolation will be different from that predicted by the simple theory, even with 'ideal' mountings. In practice, real bodies have resonance modes just like those of the anti-vibration mountings described in Section 2.3.1. In the absence of any loss components the impedance with which the mount is loaded can take all possible reactive values from $+j\infty$ to $-j\infty$. For the isolation

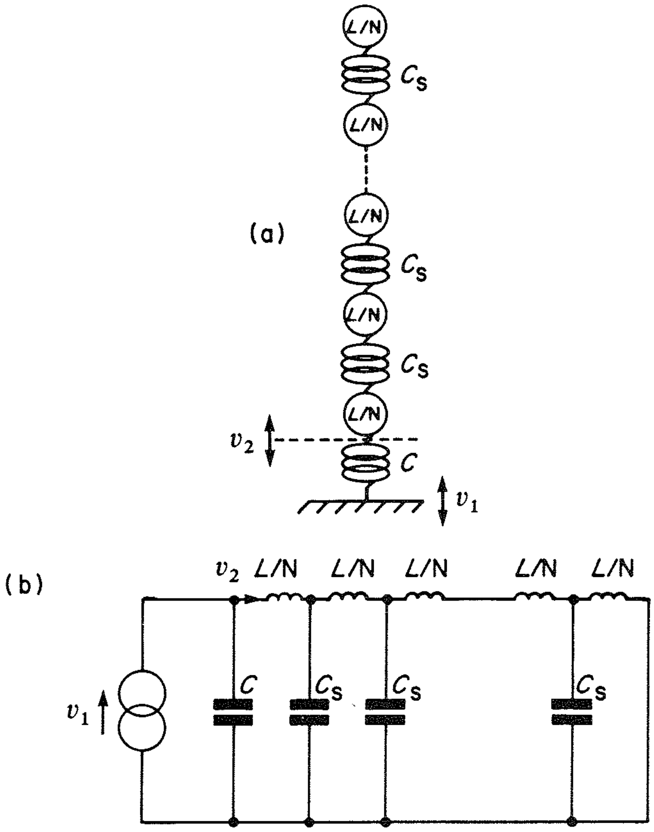

Fig. 7 – Vibration isolation system with resonance modes of the supported mass.
(a) Mechanical circuit.
(b) Electrical equivalent circuit.

of entire rooms, especially large rooms such as drama studios, the highest frequency at which the structure may be considered as a simple mass is within or below the audio frequency band, so that the whole operating range of the vibration-isolation system is in this modal regime. An example is described in Section 3.2.4.

The equivalent cicuits of an ideal mounting with a non-ideal load are shown in Fig. 7. For simplicity, the load has been assumed to comprise a number, N, of equal masses, each N times smaller than the whole mass. These masses are connected by compliances, C_s, (corresponding to the flexibility of the structure) in a simple linear arrangement. In practice, the inter-linking of the constituent masses will be three dimensional, that is if the masses are even distinct. Also for simplicity, the damping components have been omitted from the drawing (but not from the following calculations). The model is therefore very simplified but will serve to illustrate the effects. Fig. 8 shows the isolation response calculated for the same conditions as for Fig. 5 but with the mass divided up into 10 equal parts, connected by compliances, also of equal values, appropriate for modal frequencies approximately in the range 20 Hz and upwards. The assumed damping components correspond to 'Q-factors' of about 20.

Again, the results are similar in kind to those obtained from non-ideal mountings with ideal loads. Both types of imperfections cause the same shortfalls from the ideal performance at some frequencies and, in general, both types will be present in any practical construction. There are, of course, other frequencies where the isolation is improved with respect to the theoretical ideal. For the isolation of discrete machine vibrations some advantage can be taken of this, or at least no disadvantage, by choosing that the adverse modes of the springs do not correspond with any of the frequency components present. For the isolation of structures from continuous spectrum noise this choice is not available and the worst-case available isolation will always be governed by the adverse resonances in either the mounts or the structure. This marks the divergence of the two vibration isolation requirements and is discussed further in Section 2.3.4.

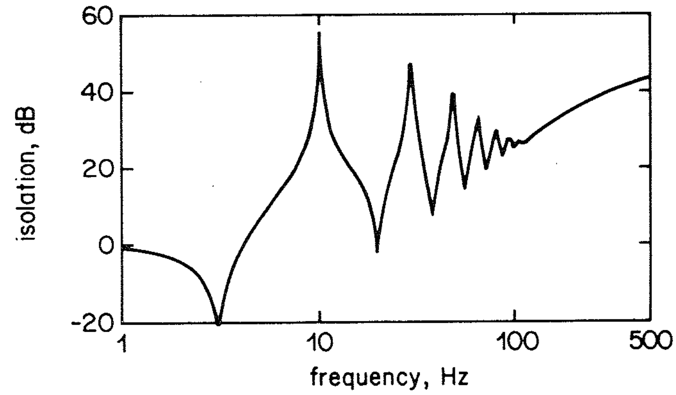

Fig. 8 – Isolation response of the circuit of Fig. 7(b).

2.3.3 Acoustic coupling

One means by which vibration energy may be coupled from a source to a receiver is acoustically, via the intervening airspace. Section 2.1 described the coupling from the surface of a structure to the adjacent layers of air and thereby to the volume of the enclosure. Reciprocally, air movement can induce mechanical vibrations into a structure. Because of the difference in densities of the two media the effectiveness of the coupling is different in the two directions.

Consider a structure comprising two plane, parallel, limp (that is, with no inherent stiffness) walls of infinite extent separated by a large airspace. If one of these walls is vibrating with a fixed magnitude then, by the same arguments as in Section 2.1, the velocities of the vibrations in the air-layers adjacent to this wall will be equal to those of the wall. If this sound energy propagates as a plane-wave, as it will if the vibration amplitude is uniform all over the wall, then it will arrive at the opposite wall surface with the same magnitude, ignoring any energy loss due to propagation (which will be very small over short distances and at moderate frequencies). The pressure exerted on the second wall by the sound wave will depend upon the acoustic pressure which is related to the acoustic velocity by the acoustic impedance. In conformation with Newton's Laws of motion, this pressure variation will then induce mechanical vibrations in the second wall. It can easily be shown that the ratio between the vibrations in the two walls, that is, the effective coupling of the vibrations, is given by:

$$\rho_0 c / m\omega$$

where $\rho_0 c$ is the plane wave acoustic impedance
m is the mass of the second wall, per square metre
and ω is the frequency in radians per second

At a frequency of 100 Hz and for a wall of mass 625 kg/m^2, this expression has a value of -59 dB. Therefore, if this arrangement were part of a vibration isolation system, there would be a theoretical limit of 59 dB for the isolation for this factor alone. The assumption of plane, parallel walls and of plane-wave propagation mean that this limit is theoretically independent of the spacing between the walls.

In practice, the walls are not infinite in extent or plane or parallel and, therefore, the propagation of the sound energy within the cavity is not so simple. But at low frequencies, with the typical dimensions of ordinary cavity walls, the idealised model would be a reasonable approximation. In Section 2.1, for the purpose of calculating the regenerated noise levels in a room as a result of surface vibrations, it was assumed that the propagation in plane-waves was true locally. It will be shown later, in Section 3.4, that this is a reasonable assumption, for that purpose, over a large part of the audio frequency spectrum. If the airspace thickness is comparable with the extent of the locally plane-wave radiator then the simplifications are no longer reasonable.

Also in practice, the vibration modes of the structure, described in 2.3.2, have an effect on the magnitude of this airborne flanking path. At some frequencies and at some positions on the walls' surfaces there will be an amplification of the vibrations, both of the 'transmitting' wall and the 'receiving' wall. If the modal 'Q'-factors of the walls were about 20 then the effective acoustic flanking path could be increased, at some frequencies and positions, from the -59 dB calculated above to -33 dB, even if there were no coincidences between the modal frequencies of the two walls. The effect of this modal amplification averaged over the whole structure will be a reduction of the acoustic coupling isolation limit. For full-size structures, where the modal density is high over most of the audio frequency range, this effect will occur at all significant frequencies. It will be shown later, in Section 3.2.3, that measured values for this limit of about 20–30 dB were encountered several times during the course of the experimental and practical work.

2.3.4 Differences between source isolation and receiver isolation

Vibration isolation systems are frequently encountered which, despite all of the above limitations, apparently work adequately. However, these are invariably installations isolating a source of vibration, usually a machine of some kind, from the supporting structure. In such cases, most of the limitations either do not apply or, effectively, are hidden. Machines generally produce vibrations with predominantly line spectra, that is, the energy only exists at certain, isolated frequencies. If the mountings do not have a resonant mode at any of those frequencies then their defects are hidden. Even better, if the mountings have anti-resonances at those frequencies then their performance will exceed that which could theoretically be expected. It is common industrial practice in such installations which are ineffective initially to change the models and the arrangements of the mountings (but usually not their types) and, perhaps, the 'noise-stop' pads until adequate performance is obtained. Structural modes in the machine or in the supporting structure can also result in inadequate performance and, in critical and expensive installations such as naval engine mountings, can be designed to be at frequencies where they do not cause problems. The acoustic coupling is not usually a serious problem because the radiating surface of the machine is relatively small and it can be surrounded by a sound-absorbing and insulating enclosure.

Sources, such as underground railway systems, which do not have the vibrational energy confined to discrete frequencies are nevertheless still easier to control at source than at some remote receiver because the vibration is localised. With suitable mountings (it will be shown later, Section 3.3, that there are some types which are less prone to modal behaviour than others) and with acoustic enclosures, the vibration can be contained to some extent[8].

In the design of vibration isolation systems for entire rooms remote from the source of vibration and for a continuous spectrum of vibration, none of these degrees of freedom is easily available. The continuous spectrum means that the isolation system must be adequate at all frequencies. The modal density of practical, full-size rooms is high or very high at all frequencies of interest and essentially incalculable because of the inevitable vagaries of building construction work. Finally, the areas across which acoustic coupling can take place are large and, usually, closely spaced. To surround the room with an insulating enclosure to protect it from the airborne noise is neither simple nor cheap. Section 7 describes some proposals whereby the additional cost of such enclosure might be reduced but it would still be significant.

2.3.5 Performance envelope of practical vibration isolation systems

The ideal, theoretical performance of a vibration isolation system can be combined with the factors discussed in Sections 2.3.1 to 2.3.3 to derive an envelope within which the performance of a practical isolation system might be expected to lie. The assumptions made are:

1) The supported load is a drama studio with a mass of 450 tonnes

2) The static deflection of the supports is 25 mm. Much more would be somewhat impractical and any less would give little

isolation at the low audio frequencies. This gives a theoretical fundamental frequency of 3.15 Hz.

3) The mounts themselves have a combined mass of 10 tonnes and are themselves supported on a structure with a static deflection of 3 mm. This might be the 'noise-stop' pads of helical steel springs or the inherent compliance of the supporting structure.

4) The 'Q'-factors of the mount modes and the structural modes are 20 and that of the fundamental resonance is 10. Below about 10 Hz the studio structure behaves as a simple mass. Below 50 Hz the mounts behave as pure compliances.

5) The structure of the studio walls and floor and ceiling can be represented by an average of surface density 625 kg/m^2, corresponding to a 250 mm thickness of dense concrete.

6) The main supporting structure completely surrounds the studio.

Fig. 9 shows the expected, practical isolation performance, derived in stages. Fig. 9(a) shows the theoretical performance of the ideal system, including only the fundamental loss component of the mountings. The irregularities near 100 Hz and the change in slope are the result of the interaction between the 'noise-stop pads' and the mass of the mounts.

Fig. 9(b) shows the combined effect of the mounting and the structure modes and is derived from a combination of the type of effects shown in Figs. 5 and 8. A 'Q'-factor of 20 corresponds to an amplification of 26 dB at a modal frequency. In the region between the fundamental resonant frequency (3.15 Hz) and the frequency of the first mode of the structure (10 Hz), the performance envelope makes a transition between the lines (a) and (b). The form of this transition is not well-determined in general and it is merely indicated here for completeness.

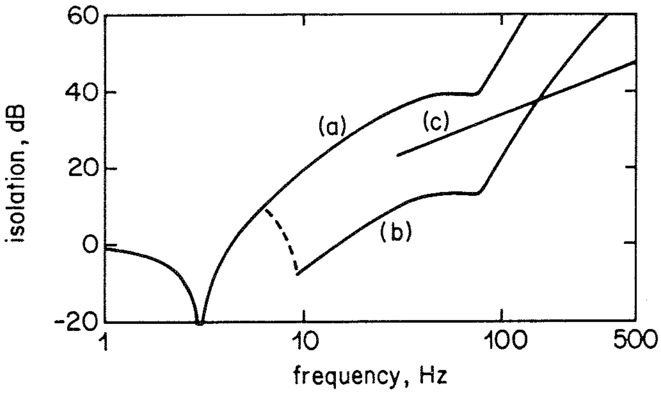

Fig. 9 – Envelope of practical achievable isolation.
(a) Ideal system.
(b) With allowance for mount and structural modes, −26 dB.
(c) Acoustic flanking path.

Finally, Fig. 9(c) shows the calculated limit due to the acoustic coupling between the supporting structure and the studio, following the arguments of Section 2.3.3.

Because the practical limits to achieved isolation performance occur as a result of the various resonance modes, the whole of the isolation characteristic will not be at the levels shown. Rather, the lines in Figs. 9(b) and 9(c) represent the loci of all possible points of the minima in the isolation characteristics.

3. THE LANGHAM TEST SLAB AND OTHER MEASUREMENTS

3.1 Vibration measurement – introduction and general considerations

In general, measurements of vibration are made using a type of transducer which is sensitive to the acceleration of the surface to which it is attached. Apart from limitations inherent in the transducer itself, such as an internal noise level below which no measurements can be made, there are also external limitations. The 'accelerometer' has a mass of its own (in fact, most of them work by measuring the force required to accelerate an internal mass) so that, for the measured results to be an accurate representation of the movement of the surface on which it is mounted, the transducer must have a small enough mass to have a negligible influence on the movement of the surface. It must also be firmly connected to the surface because any compliance in the mounting will behave as an 'anti-vibration' mounting and will isolate the transducer from the movement which it is supposed to be measuring. Having a small mass helps in this respect also. However, a small mass also makes the transducer insensitive. These requirements are clearly in conflict although, for the measurement of vibration in large concrete structures, the mass of most reasonable accelerometers is not sufficient to affect the vibration of the surface significantly.

Except for some early measurements of site vibrations at the foundations of the existing Langham building and for some experiments in the design of very sensitive accelerometers (see Section 3.7) most of the measurements carried out during the course of the work were made with Bruel and Kjaer, type 8306 accelerometers, which are the biggest and most sensitive readily available. They have a frequency response extending up to 1 kHz which was entirely adequate for the purposes of the measurements. For most of the measurements described in this report the accelerometers were secured to the measurement points by a threaded stud connection to steel adaptor plates which were themselves secured to the structure by epoxy adhesive. In addition to providing very rigid connections for the measurement points, this simplified the use of the same points for different measurements and also meant that the points could be re-used if it was required to repeat any measurements.

For some of the measurements in existing buildings, for example in the corridors and stairwells of Broadcasting House, the accelerometers were secured to the measurement points by plastic modelling clay after the surface had been cleaned free from dust. This may appear to be a less than secure method of attachment after the comments made above but this material is inert (that is, it has no discernible resonances), it conforms to surfaces, it has a high bulk modulus and it is sufficiently adhesive to withstand the dynamic forces involved at the low levels of vibration normally encountered in this type of work. With the 8306 accelerometer, it has been found to be satisfactory, even in quite thick (1–2 mm) sheets, up to 1 kHz.

3.2 The Langham Test Slab[9,10]

3.2.1 Description

As outlined in Section 1, the main part of this work on vibration isolation was concerned with measurements on a structure intended to represent the behaviour, on a smaller scale, of a drama studio on the Langham site. It was constructed in the basement of the existing Langham building and at a distance from the nearest Underground train tunnel which was representative of the distance at which a drama studio might eventually be located. At the point of nearest approach the rail tunnel was about 18 m laterally and 18 m vertically from the edge of the Test Slab. There was, therefore, no difficulty in obtaining adequate signals from the vibration transducers and in having available sufficient margin of signal-to-noise ratio to permit the measurement of high values of vibration isolation.

Although eventually the construction was extended to model a complete room, most of the fundamental measurements were carried out on an essentially two-dimensional (flat) structure. It consisted of a heavily reinforced concrete slab, 125 mm thick, measuring about 3.6 m × 5.9 m. The edges were stiffened by an upstanding ring-beam which was 400 mm high and 215 mm wide on the two long edges and one of the short edges. On the fourth edge the upstand was 225 mm high and 1000 mm wide. This reduction in the beam height was intended to accommodate the doorway and to facilitate access at the later stages of development when the complete room was finished. The bottom surface was flat and unobstructed so that the airspace underneath was well-defined, for the purposes of investigating the effects of varying the airspace. Three hydraulic jacks and associated steelwork were attached to the slab with sufficient total lift to give an airspace height of 1000 mm at their maximum extension.

The foundation slab for this construction was about 600 mm thick, also heavily reinforced and supported in turn on nine piles passing through the floor surface to the main site-slab for the building which was about 1500 mm further down. It had been established that the site-slab was about 1000 mm thick and had been in position for over 100 years. The supporting structure was therefore very rigid and well-connected to the underlying strata and thus to the ground vibrations.

The Test Slab weighed about 11 tonnes and two independent safety systems were incorporated, either of which would support the slab after a fall of 50 mm. A system of variable-height interlocking steel stools was provided for setting the height of the slab, on a set of anti-vibration mountings, to any value up to 1 m.

Fig. 10 shows a general view of the arrangement of the slab and its supports.

3.2.2. Measurements – vibration isolation

Measurements of vibration isolation of the Test Slab supported on a set of vibration isolation mountings were made using a pair of accelerometers, one measuring the vibration levels in the foundation slab resulting from the passage of an underground train and the other mounted on the Test Slab. With such large structures the pattern of vibrations is complicated at any frequency above that of the lowest resonant mode; in particular, at any point the vibrations may occur in the directions of all six degrees of freedom of a free mass. It was not practicable or especially meaningful to measure all of these vibrations individually. Of course, in the final structure it is those components of vibration which take place normal to a surface which regenerate most of the acoustic noise. However, the modal nature of the vibrations and the three-dimensional nature of a real structure mean that all vibration modes will have a component normal to the surface at some point. For the purposes of comparison, all of these measurements of vibration isolation were carried out using the vertical translation component only. Some check measurements of the other translation components showed that they were similar in kind, although they did give some detailed differences. (It will be seen shortly that the details of the vibration isolation measurements are of less importance than their general order.) No attempts were made to measure any of the rotational components but there is no reason to suppose that they were any different in kind.

The signals from the accelerometers were processed by a two-channel 'Fourier Transform Analyser'. This instrument carries out a discrete (quantised) version of the Fourier Transform to obtain frequency-domain spectral information from a time-domain input signal. Having two channels, it can directly calculate and display the ratio between the two inputs in amplitude and phase as a 'Transfer Function'. This can be interpreted directly as the transmissibility of the system, the reciprocal of isolation. For consistency, all of the discussions and results for the ratios of vibrations presented in this report have been converted to 'isolation', whatever the basis of the original figures. For the purposes of isolating vibrations with a continuous spectrum it is the lowest value of isolation obtained over a range of frequencies which is important. It is of little value if an isolation system is effective at all frequencies but one. In practice, it will be seen later (Section 5) that the isolation required for a drama studio is a function of frequency so that comparisons of the achieved isolation with the requirement is not simple, but it is still the minima of isolation which are important; the maxima are irrelevant.

Measurements of vibration isolation were carried out on seven different types of mounting chosen by the Acoustic Consultant. Unfortunately, the results from the different types of mounting are not directly comparable because the mounts were not all designed to have the same static deflection and hence the same resonant frequency and idealised isolation performance under the load of the Test Slab. The types of mountings and brief descriptions of their constructions, advantages and forseeable problems are given in Appendix 3.

In addition to the different types of mountings, some other variables were also included in the tests. The effect of the height of the airspace beneath the supported slab was expected to be a significant parameter because of the acoustic behaviour of the air in the space between two surfaces. So, for a number of different arrangements of mountings, the vibration isolation was measured at at least two different heights. For the same reasons, several measurements were made with glass-fibre mat filling part or all of the height of the airspace.

In all, 16 different conditions had to be measured over a ten-week period. The vibration isolation was measured for each condition at six different pairs of positions, and plotted as a function of frequency. It is customary for acoustic data to be plotted in 1/3rd octave bands, a form of display which usually corresponds well with the subjective significance of the data being presented. It became clear at the outset, however, that if the objective of the work was to understand the physics of the systems, then the data would have to be displayed in narrow-band form. An example of the 1/3rd octave plot is shown in Fig. 11 for four airmounts, and the same data in narrow-band form is presented in

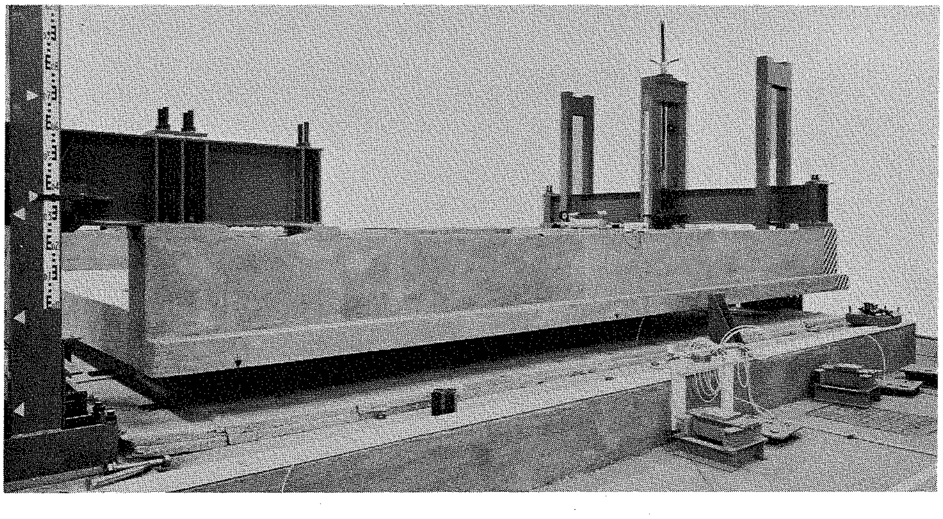

Fig. 10 – The Langham Test Slab.

Fig. 12, which shows several interesting features not apparent in the 1/3rd octave plot. The Fourier Transform analyser used for the measurements generates its initial data in narrow-band form, using further processing to produce 1/3rd octave plots if required; in other words, all of the data collected is in any case available in both forms.

Also for each condition, the vibration isolation function was measured using the narrow-band frequency analysis over the frequency range, 0–500 Hz at one, or sometimes two, positions. Figs. 12, 13, 14 and 15 show samples of the results obtained. Also shown on those figures as a dashed line is the theoretical isolation obtained from the simple, three element equivalent circuit with the appropriate fundamental resonant frequency. The overall impression to be obtained from these sample results is that they are not much like the theoretical except for the region near to the fundamental resonant frequency. In all cases, the functions are characterised by large and irregular variations with severe shortfalls at some frequencies. Some of these

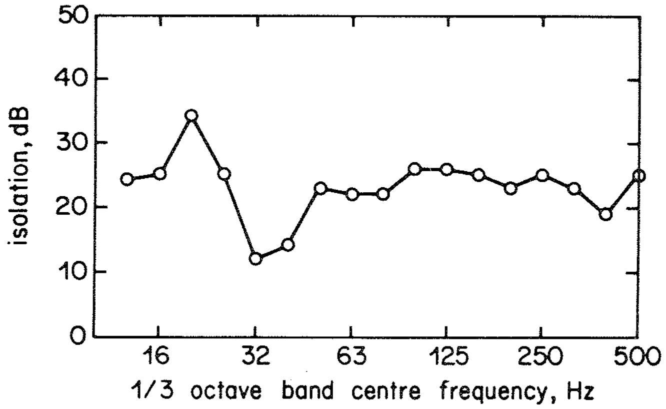

Fig. 11 – Measured isolation in 1/3rd octaves, four airmounts.

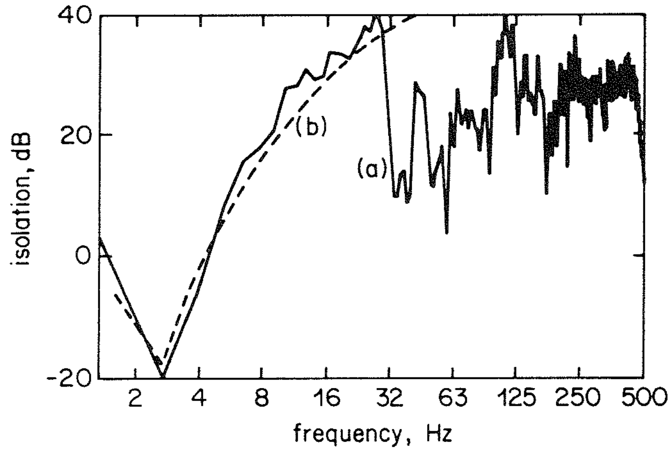

Fig. 12 – Measured narrow-band isolation, four airmounts.
(a) Measured results.
(b) Prediction from simple model.

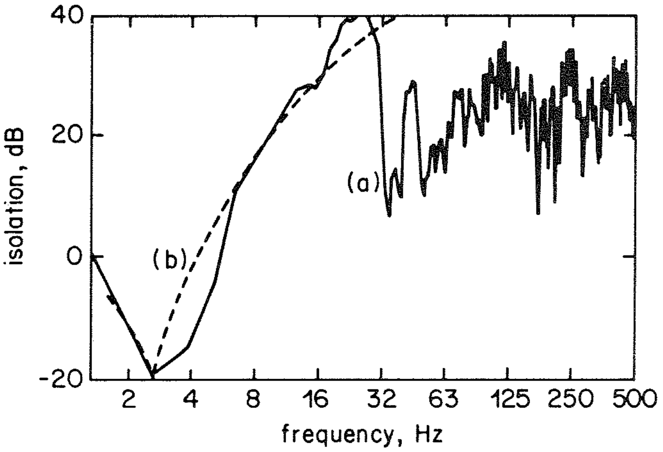

Fig. 13 – Measured narrow band isolation, four helical steel springs.
(a) Measured results.
(b) Prediction from simple model.

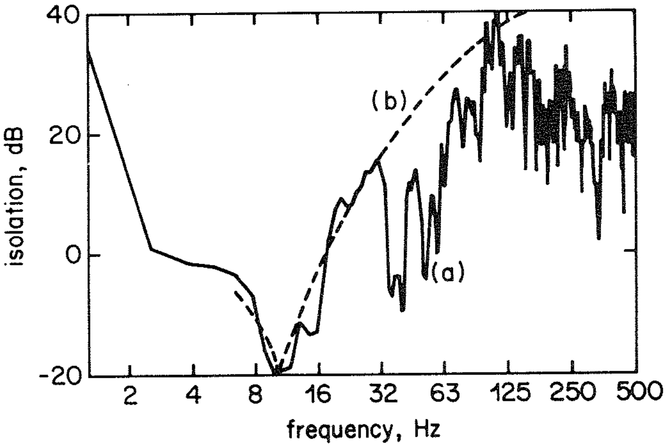

Fig. 14 – Measured narrow band isolation, four rubber compression mounts.
(a) Measured results.
(b) Prediction from simple model.

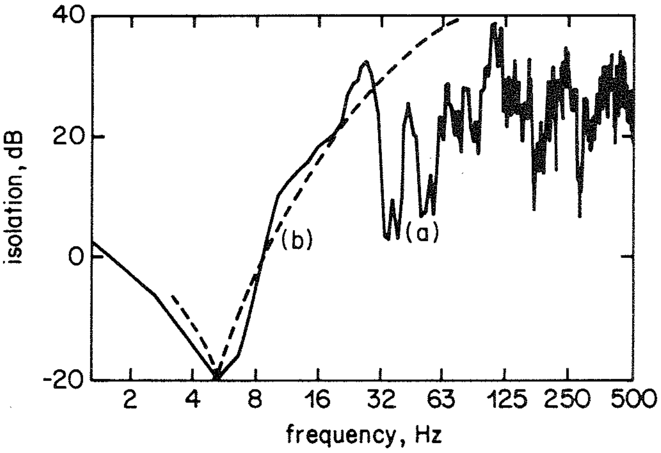

Fig. 15 – Measured narrow band isolation, four rubber shear mounts.
(a) Measured results.
(b) Prediction from simple model.

shortfalls, particularly the pairs at about 35 Hz and 50 Hz and the group at about 130 Hz, appear in all of the results. The 35 Hz frequencies were clearly shown to be those of at least two major modes of vibration of the slab (see Section 3.2.4). The apparently worse performance of the rubber mount at low frequencies is entirely due to the inappropriate stiffness, resulting in a much higher fundamental resonant frequency than for the other mounts. In other words, as stated previously, the measured isolations for the different mount types should not be directly compared with each other; instead, each should be compared with the corresponding simple model whose isolation appears on the same figure.

The similarity of form between these sample results and those derived theoretically and shown in Figs. 5 and 8 was encouraging. Although the real structure had many more modes than it was possible to model reasonably and their frequency distributions were very different, it is evident that the mechanisms have been identified and modelled. The 'performance envelope' derived in Section 2.3.5 and shown in Fig. 9 is, in principle, a reasonable description of the measured behaviour of all these types of mounts, with due allowance for the differences in fundamental resonant frequencies.

The results shown in Fig. 12 for the airmounts are for the same condition and measurement position as for the 1/3rd octave results shown in Fig. 11. The reduction in the information content of the wider band analysis is evident.

The effect of changes in the height of the airspace beneath the slab is illustrated by Fig. 16. The measured isolation for the arrangement of four airmounts with total airspace heights of 230 mm and 600 mm are shown. The airmounts were chosen for this comparison because in giving relatively high values of isolation with fewer unwanted modes, the effects of changes in the airspace height should be more distinguishable. Fig. 16 shows that the effects are very small. The differences are very much less than those between the results for different measurement positions for the same support conditions.

The effect of filling the airspace under the Test Slab with glass-fibre mat is illustrated by Fig. 17 which shows the measured isolation using the small glass-fibre vibration isolation cubes with (a) and without (b) the remaining space being filled with low-density mat. Three general changes appear to have resulted from the insertion of the mat; the 'Q'-factor of the fundamental resonance is lowered (measured separately, from about 16 to about 5), there is a redistribution of the modes between 30 and 70 Hz and the minima in the isolation above about 100 Hz are improved by an average of about 7 dB. These results are discussed further in Section 3.2.4. In this case, the small dimensions of the cubes meant that the airspace could be completely filled with the glass-fibre mat. Other tests, on larger mounts but with the same thickness of mat, showed that filling only a relatively small fraction of the airspace height with the mat was completely ineffective.

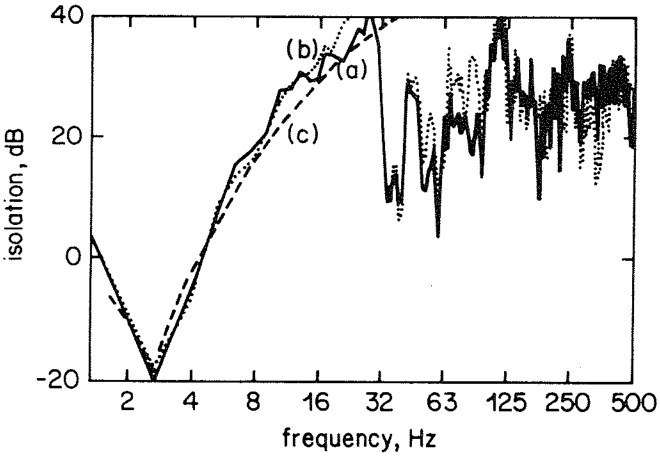

Fig. 16 – Measured narrow band isolation, effect of altering the airspace height.
(a) 230 mm.
(b) 600 mm.
(c) Prediction from simple model.

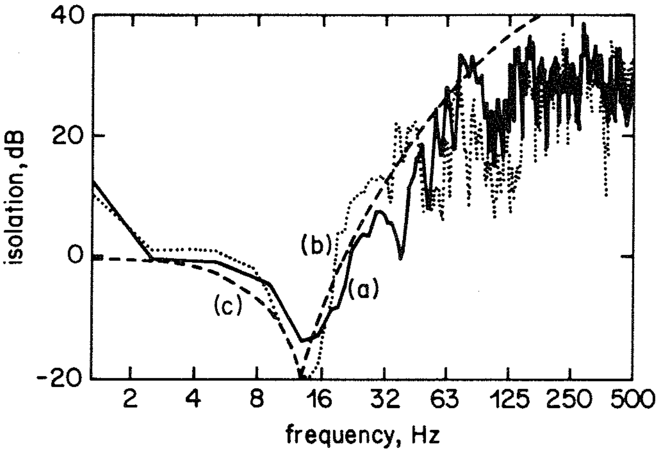

Fig. 17 – Measured narrow band isolation, small glass-fibre cubes.
(a) With glass-fibre infill.
(b) Without glass-fibre infill.
(c) Prediction from simple model.

3.2.3 Measurements – airborne coupling

It was always evident that the Test Slab would be affected by the acoustic noise generated in the room in which it was built. However, apart from the theoretical considerations described in Section 2.3.3, there were no preconceptions about the relative importance of that contribution. As soon as the earliest measured isolation results were obtained it became evident that some process was limiting the achieved minima of isolation to about 15–20 dB at all frequencies above about 40 Hz, even with the types of mounting which were expected to be inherently relatively free from unwanted modes. This effect can be seen clearly in Figs. 12–15.

As both the acoustic noise and the ground vibrations were generated by the passage of the trains, it was not possible to separate their effects at the time of the measurement. It was, however, possible to estimate the magnitude of this effect from independent measurements of vibrations generated by trains and acoustic noise generated by a loudspeaker. Two assumptions are implicit in this estimation. The first that the acoustic noise generated no vibrations in the foundation slab; this is likely to be a reasonable assumption because of the great mass and stiffness of that foundation. The second is that the distribution of the acoustic energy was similar whether it was caused by a train or the loudspeaker; except for the fairly confined space beneath the Test Slab, this is also likely to be true because the room was large and reverberant and two distant, spaced loudspeakers were used.

The principle of the method was that the vibrations in the foundation, A_t, and the sound pressure level near the Test Slab, P_t, caused by the passage of a train and the vibrations in the supported Test Slab, A_s, and the sound pressure level near the Test Slab, P_s, caused by the loudspeaker should be measured. The vibration caused in the Test Slab by the acoustic noise generated by the train A_{st}, is then given by:

$$A_{st} = A_s . P_t / P_s .$$

The maximum measurable isolation is then given by:

$$A_t / A_{st} = (A_t / P_t) / (A_s / P_s)$$

The function A/P was measured as a transfer function by the signal analyser twice, once for the passage of a train and once with acoustic noise radiated by the loudspeaker. The ratio of the two transfer functions gave the isolation limit set by the acoustic flanking path. One advantage of this method is that all of the transducer sensitivities and calibrations cancel out, leaving only the wanted function.

Fig. 18 shows the result obtained using narrow-band frequency analysis. Results were not obtained below 30 Hz because of loudspeaker limitations. Although the result is highly irregular and shows the same sensitivity to vibration modes in the Test Slab as do the isolation results, it is evident that there are several minima reaching 15 dB and many reaching 25 dB. At those frequencies the acoustic coupling between the room surfaces and the Test Slab limits the achievable and the maximum measurable vibration isolations to those values. Fig. 19 shows the same results, converted to 1/3rd octave analysis. Much of the detail has disappeared but it is even more clear that the acoustic coupling is an important limiting factor. In Fig. 20 the measured acoustic coupling limit, in 1/3rd octave bands, is compared with the measured vibration isolation for the same system of mounts, also in 1/3rd octave bands. Clearly, the acoustic coupling is entirely responsible for the measured values of isolation, at all frequencies above 25 Hz (and that frequency may be a limit due to the measurement technique) and is limiting the maximum isolation obtainable using the Test Slab to between 12 and 29 dB. These results also show the obvious inadvisability of attempting to measure the differences between arrangements of mountings in the presence of such a dominant flanking path.

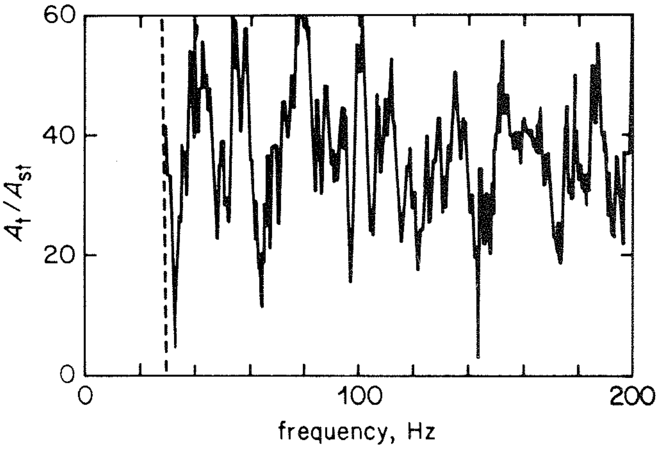

Fig. 18 – Acoustic flanking limit, narrow-band measurement.

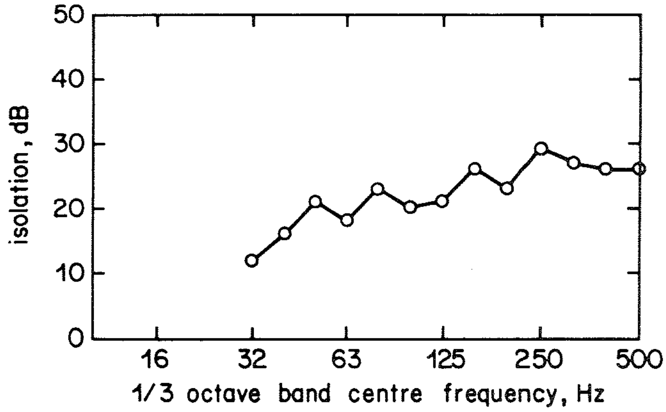

Fig. 19 – Acoustic flanking limit, 1/3rd octave measurement.

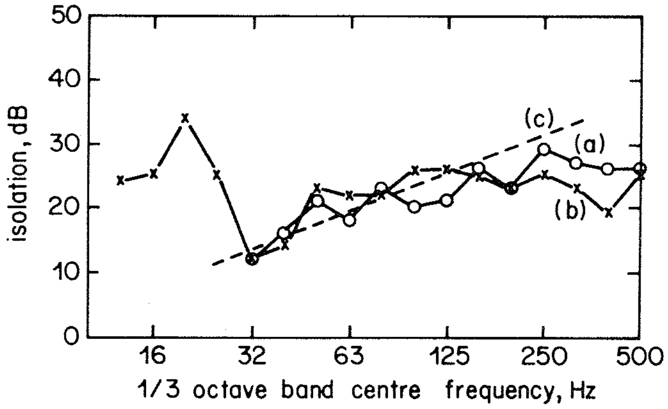

Fig. 20 – Acoustic flanking limit, measured in 1/3rd octaves, compared with measured vibration isolation.
(a) Acoustic flanking limit.
(b) Measured vibration isolation.
(c) +6 dB/octave.

The simple theory of Section 2.3.3 indicated that the isolation limit due to the acoustic coupling should increase at a rate of 6 dB/octave. In the results shown in Fig. 18 there is no substantial indication that this slope is present. Figs. 19 and 20 do indicate an average slope of about that value for the lower end of the frequency range. Figure 20(c) is a line of slope 6 dB/octave at an arbitrary position. However it is obvious that the practical limit is not directly defined by the mass of the structure, otherwise it would be about 54 dB at 100 Hz (see Section 2.3.3). It is clear that the amplification effects of the structural modes are, as expected, responsible for very significant reductions from the values predicted from the simple, limp-wall theory.

3.2.4 Measurements – slab modes and 'Q'-factors

It is by now evident that the achievable vibration isolation of a large structure is heavily dependent on its modal behaviour. Both the performance of the vibration isolating mounts and the magnitude of the effective acoustic flanking are influenced at and near to the modal frequencies. It was realised that it would be impossible to design a large structure with no modes within the low-frequency part of the audio spectrum where the train vibrations were a potential problem or even, because of the normal tolerances of building construction, to control the frequencies of the modes which would inevitably occur. It was nevertheless considered worthwhile to investigate the modal behaviour of the Test Slab. To do this rigorously at many frequencies would have required specialist modal analysis software for the Fourier Transform analyser. This was not considered worthwhile for the limited amount of use to which it would be put. Instead, a few simple, low-frequency modes were identified and their vibration distributions derived manually from measurements on a moderately closely-spaced (5 × 5) array of points at the resonant frequencies. Fig. 21 shows what are thought to be the first four modes of the Slab, including two modes near to 35 Hz, a frequency at which many features have shown up in other measurements. Also shown in Fig. 21 are the measured 'Q'-factors of these modes, which are about 80–90.

Estimates of the frequencies of all of the modes in the frequency range up to 200 Hz were made for all of the 16 mounting conditions by recording the transfer function between two well-spaced points on the Slab in both amplitude and phase. The intellectual basis for this measurement was that, for any mode, both of the accelerometers were unlikely to be at points of equal magnitude and phase on the mode. In general then, the Transfer Function should show a local maximum or minimum and a 180° change in the phase at the modal frequency. It later became evident that a few modes would not be identified by this measurement, but by then it was too late as most of the work had been completed. In practice, the measurement had always been repeated at several pairs of points so that the probability of a mode not showing in any of the results was small. The source of the excitation for this measurement was a repeated impact on the Slab. Fig. 22 shows a sample result from such a measurement. The 360° jumps in the phase due to the ambiguity in the cyclically repetitive phase analysis have been 'unravelled' to avoid unnecessary complication. Thus the phase axis includes several complete rotations of phase.

After the completion of the first series of measurements on the Slab, it was decided that it was necessary to add some form of damping to reduce the effects of the modes. The only practicable

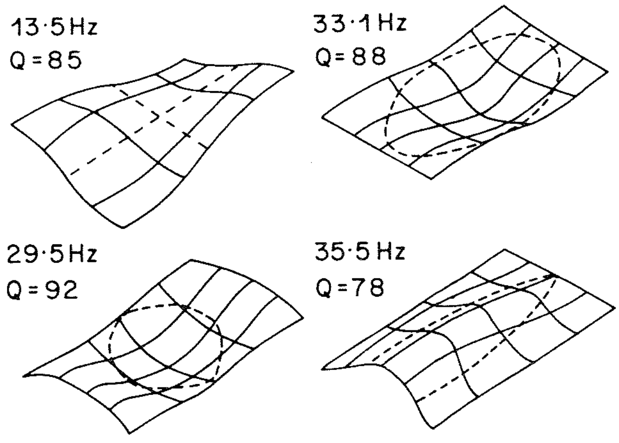

Fig. 21 – First four modes of vibration of the Test Slab.

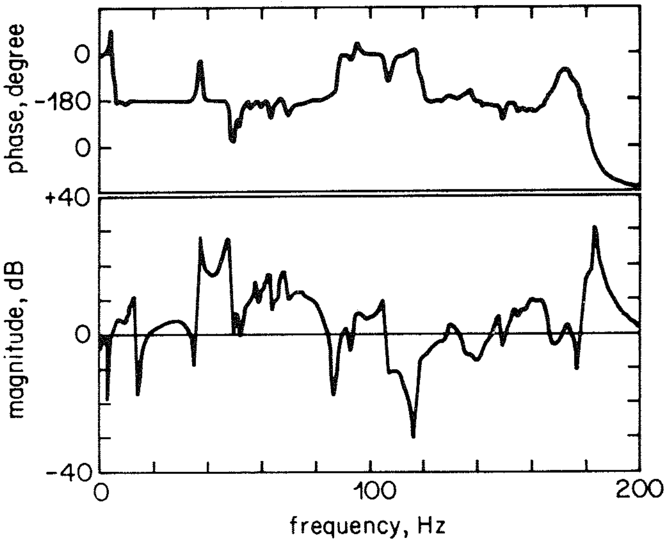

Fig. 22 – Point-to-point transfer function on the Test Slab.

method of adding damping to such large and massive structures which will be effective at all frequencies is to incorporate a 'constrained layer' of damping material. This works by having a thin sheet of material with a high damping coefficient constrained by being sandwiched between two sheets of much stiffer material. The damping material is then forced to undergo large shearing deformations for relatively small bending deformations of the composite slab. With proper control of the shear modulus of the damping layer in relation to the Young's Modulus of the constraining layers and of their relative thicknesses, very effective damping can be obtained.

Although the authors were not privy to the design process, the material used did not appear to be appropriate and it was constrained by only a relatively thin, lightly reinforced concrete screed on the top whereas it was constrained by the much thicker and heavily reinforced Test Slab on the undersurface. Many measurements and much analysis of the 'effectiveness' of this damping layer were carried out several times during the four-week period in which the new concrete was curing. In the end, no nett measurable differences in 'Q'-factors of the Slab modes were found, although during the curing the losses were slightly higher. The progress of about 50 identifiable modes were followed through the changes in frequency and loss-factors which occurred during the curing period without any significant permanent changes being observed.

The largest damping effect observed on the slab occurred when the airspace beneath it was small (about 40 mm high) and completely filled with glass-fibre mat (Fig. 17). The damping of many of the modes was notably high, with 'Q'-factors of about 20 or less (one of less than 10) for frequencies up to about 50 Hz. Above that frequency there was no consistent change. It is thought that the damping mechanism in this case was one of 'pneumatic pumping'; in attempting to move differentially, the slab caused air to be pumped through the long and narrow passage filled solidly with glass-fibre mat. The glass-fibre itself, in compression, is not very lossy – it behaves as a good quality spring. However, acoustically, it generates high losses. At higher frequencies, that is, shorter modal wavelengths, this pumping mechanism becomes less effective because of the reduction in the length of the path, thus accounting for the observed upper frequency limit.

3.3 'Ideal mass' avm measurements

The Test Slab and its associated supports constitute a very complex mechanical system. The measurement of the behaviour of one part of the system, the slab itself, has been considered (more or less) alone and some of its modes of vibration identified and measured. Another part of the system, the anti-vibration mountings, can also be investigated separately from the rest of the system. If it were possible to use a mount to support a load which had no properties other than its own mass then a measurement of the vibration isolation performance of the resulting assembly would allow the behaviour of the mount alone to be extracted. The principle is identically equivalent to loading an unknown electrical source with a perfect, known load impedance in order to determine the 'output impedance' of the source. (A more complex, matrix approach to the characterization of avms, using accelerometers and force transducers to eliminate the effects of the supporting and supported structures is given in Reference 1, page 1262 *et seq.* This method is equivalent to that described here but is, in principle, also capable of measuring the avm properties under significant loadings.)

Unfortunately, it is not possible to provide such 'ideal masses' on the scale and over the frequency range commensurate with a full-size drama studio. However, with some limitations it is possible to approach the ideal. The mass was required to be compact and stiff so that the dimensions would be small compared to the wavelength of the vibrations, up to a reasonable upper frequency limit. It also had to be heavy so as to simulate as nearly as possible the working conditions of the mounts to be tested, but not so heavy as to be difficult to move or particularly dangerous when freely supported by any of the mounts. The nearest approach readily available on the site was a stack of steel plates intended for use as spacers for adjusting the height of the Slab. Six such plates weighed 50 kg and, when bolted together, formed the heaviest hard object which could be handled safely on hard surfaces. If such an object did not behave ideally when supported on a real anti-vibration mounting then, clearly, it would not be obvious which of the components was at fault. However, ideal behaviour could only reasonably be attributed to both components individually.

Measurements were made of the vibration isolation of this mass on a number of the test mounts. Fig. 23 shows the results obtained using one of the smaller rubber compression mounts. For the purposes of comparison the inherent gradient of the isolation characteristic has been cancelled in the presentation by a calculated −12 dB/octave. Also shown is the horizontal dashed straight line representing the idealised behaviour of a system composed of perfect components with the same fundamental resonant frequency, at frequencies away from the resonance. Clearly, the test mass and compression mount system gives nearly perfect behaviour, within about 5 dB, up to about 470 Hz (cf. Reference 1, page 1255). It is improbable that this behaviour could occur from a combination of seriously imperfect components. Therefore the mass did behave (nearly) as a pure mass element up to at least 470 Hz and could be used to determine the inherent performance of the different types of mount, up to that frequency. One of the problems of the method is that it does not load the mounts to a large fraction of their design load; for the large steel springs the loading was only 2%. However, with the exception of the airmounts, the test mounts were all close to being linear systems. Even with the rubber shear mounts which showed definite stiffening as they were compressed, the non-linearity was not of a type which would radically alter their modal behaviour or damping factor. The airmounts are quite different; their principal stiffness depends on the air-pressure which, in turn depends on the intended load. At low loads and pressures the inherent stiffness and modal behaviour of their side-walls becomes dominant. Tests on this type of mount at reduced load are therefore not meaningful. Also implicit in this measurement is a measurement of the dynamic compliance of the mount at different frequencies rather than statically or just at the frequency of the fundamental resonance. For some types of mounts, particularly those using some form of rubber compound, the compliance at frequencies of a few hundred hertz might be significantly less than that measured at lower frequencies. The material moduli of rubber-like materials are usually frequency-dependent.

Figs. 24, 25 and 26 show the results obtained from this test on the rubber shear mount, the large helical steel spring mount and the glass-fibre cube

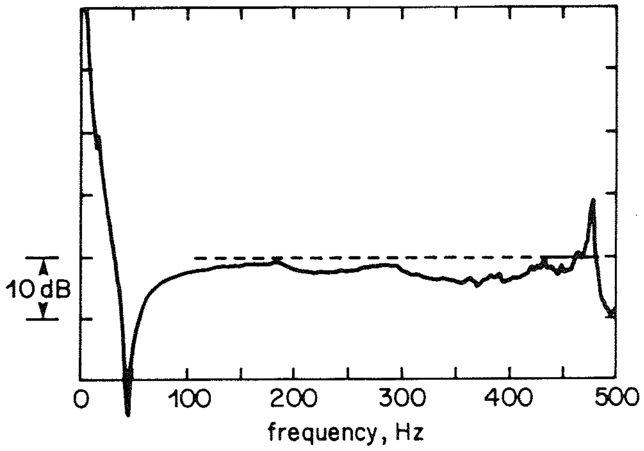

Fig. 23 – Isolation, measured using 'ideal mass', for rubber compression mounting, corrected by 12 dB/octave.

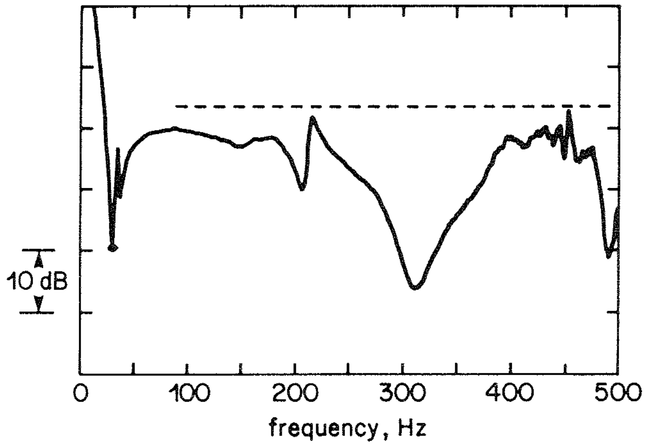

Fig. 24 – Isolation, measured using 'ideal mass', for rubber shear mounting, corrected by 12 db/octave.

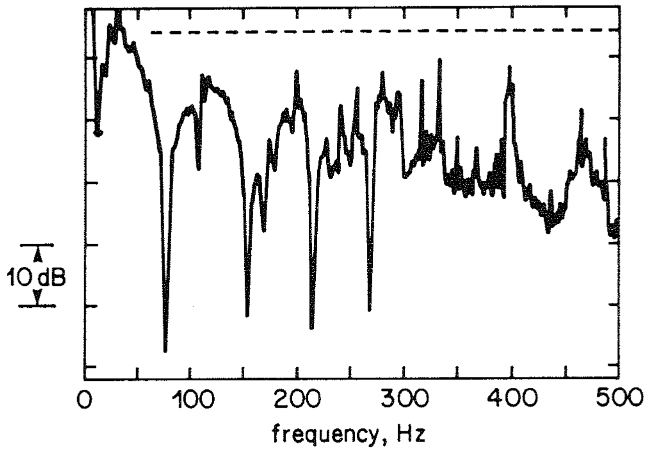

Fig. 25 – Isolation, measured using 'ideal mass', for helical steel spring mounting, corrected by 12 dB/octave.

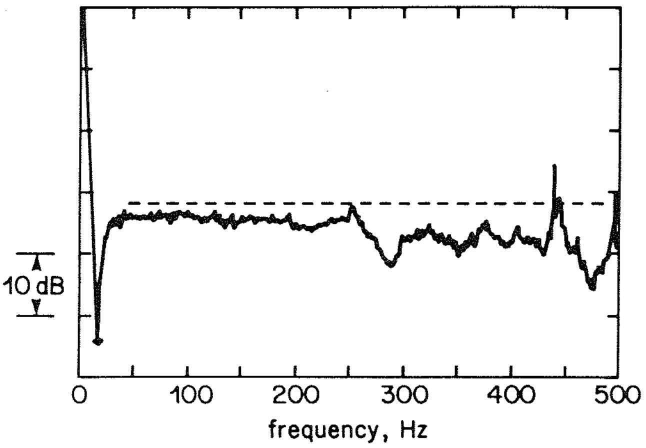

Fig. 26 – Isolation, measured using 'ideal mass', for small glass-fibre cubes, corrected by 12 dB/octave.

mount, again with the line representing ideal behaviour. The shear mount had large steel plates attached which were impossible to damp or to prevent from vibrating. They may have been responsible for the large deviations between 200 and 400 Hz. The detail of the results changed significantly for small changes in the position of the test mass. In practice, these plates would be firmly connected to the concrete of the structure.

The results for the large steel spring dramatically show the modal behaviour inherent in this type of mount. The line representing the ideal behaviour of this system is only just within the dynamic range of the measurement shown by the scale in Fig. 25. Departures of more than 50 dB from the ideal are shown, with major modes at about seven different frequencies, all within the low-frequency end of the audio spectrum (cf. Reference 1, page 1255). This measurement did include the 'noise-stop' pads supplied by the manufacturer to control the high-frequency modes. In fact, steel helical springs, in having a transmission path which is long and thin (when measured round the coils rather than end-to-end over the coils) and being made of a dense material, are probably the worst possible arrangement of that type of material for the purposes of isolating wide-band vibrations (see Section 2.3.4).

The results for the small glass-fibre cubes had been expected to show no departures from the ideal because of their light weight and small size. However, the cube is so small that the test mass overhung more than for the other mounts and may have been showing a bending mode not evident in the other tests.

Overall, the results are much as expected. The types of mount involving small dimensions and short vibration conduction paths are relatively free from modal effects, at least up to a few hundred hertz. In particular, those compression mounts which were tested were completely free up to the limit of the measurement. Much larger compression mounts which would be required to support a full-size studio would not be so free from modal behaviour but the number of modes should be small and they should also be relatively well-damped. The helical steel springs showed very pronounced modal behaviour, even at relatively low frequencies. Much larger springs which might be necessary to support the greater load of a full-size studio, weighing not 11 tonnes but perhaps 500 tonnes, would have modes at even lower frequencies.

It is interesting to note that all of the mount types, even those with no discrete modes, give less isolation than would be expected from the resonant frequency. In the cases of those with relatively uniform responses, the difference is consistent and between 3 and 5 dB. For lack of time, the reasons for this shortfall were not investigated, but they are most probably as a result of the changes in dynamic compliance with frequency.

3.4 Surface-to-air coupling

In Section 2.1 assumptions were made about the way in which mechanical vibrations of a surface are coupled to the adjacent air layers and, thence, to the body of a room as acoustic noise. The effectiveness of this conversion is one of the three principal factors in the derivation of an adequate performance target for a vibration isolation system, the other two being the level of vibrations at the source and the desired background noise level in the room.

The practical conversion factors as functions of frequency were measured in an existing 'typical' drama studio. The studio chosen for the measurements was B11 in Broadcasting House Extension, London. It is a relatively large studio (14.6 × 9.5 × 4.0 m) and, although it is supported on a rubber vibration isolation system, suffers from significant vibration interference from the Bakerloo Line of the Underground system. The acoustic treatment and the reverberation times are typical of drama studios generally.

Using the two-channel facility of the Fourier Transform Analyser it was possible to measure directly, as a function of frequency, the ratio between the sound pressure level at one point in the studio and the vibration acceleration normal to the surface at a point on the studio boundary surfaces generated by the passage of a train. This ratio can be expressed, and indeed was measured, as a transfer function, spl re. 20 μPa/acceleration re. 1 ms^{-2}.

In all, 13 accelerometer positions distributed on the walls, floor and ceiling and 3 microphone positions, spaced thoughout the volume of the studio, were used. For each combination of microphone and accelerometer positions the transfer function was measured during the passage of a train, a total of 39 measurements. The signal analysis was carried out in 1/3rd octave bands over the frequency range 12.5–500 Hz.

The results obtained consist of 39 sets of 17 transfer coefficients and are too many to include here in full. The consistency of the results was reasonably high, considering the possible number of variables involved; few results differed from the average for a frequency band by more than 10 dB and most were within 3 dB. Many data-processing strategies were considered. It could be argued that isolated, high values of transfer function might arise because, at that particular accelerometer point, the vibration levels were small and therefore not significantly influencing the measured sound pressure level. It could equally be argued that that point might be a particularly efficient radiator and that the value measured was genuine. There is no way, short of a thorough sound intensity survey, of distinguishing between these two alternatives.

The simplest data processing which might be used to average arithmetically all of the 39 results for each frequency band into a single, overall figure. It might also be argued that an arithmetic average is inappropriate for a variable expressed in decibels but, with so many similar elements included in the averaging, the difference would be negligible (<0.05 dB). There is also a philosophical argument about whether the results represent statistically variable samples of a constant function or true samples of a variable function. It would also be possible to exclude from the average the highest and lowest values but it would make very little difference to the results. There is, in any case, no good reason for excluding the extreme values (see above). Therefore, all of the data were averaged to obtain, for each 1/3rd octave frequency band, a single figure representing the average conversion of vibration acceleration to sound pressure level.

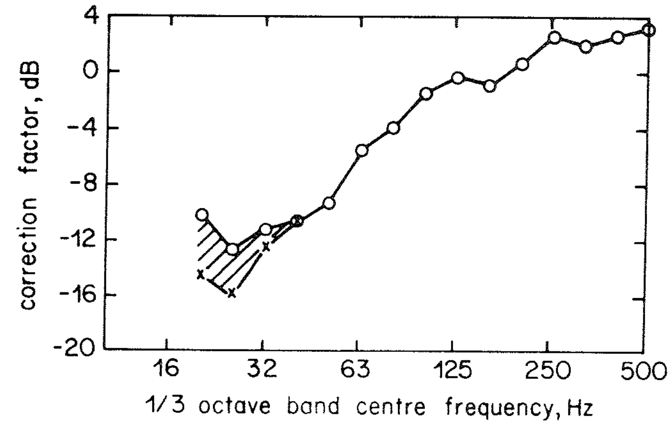

Fig. 27 – Vibration level to sound pressure level conversion: difference between measured and theoretical.

Using the theory and assumptions outlined in Section 2.1, the dimensions of Studio B11 and values of reverberation time measured for this purpose, the theoretical conversion ratio was also calculated. At frequencies below 50 Hz the reverberation time was not measured and a range of estimates was used. Fig. 27 shows the difference between the average measured and the theoretical values for the vibration to sound pressure level transfer function. The values of Fig. 27 are corrections to be added to the theoretical regenerated sound level to compensate for the observed conversion efficiencies. At frequencies from 100 Hz up to the limit of the measurement, the difference between the sound levels predicted using the simple theory and those measured in practice is less than about 3 dB with a slight tendency for the difference to increase with frequency. This shows that the assumptions made in Section 2.1 are reasonable for this frequency range. Below 100 Hz there is a pronounced tendency for the error to increase, up to a maximum of about 10 dB, at lower frequencies, as the wavelength becomes long and the initial assumptions progressively less accurate. Below 50 Hz a range of corrections is given, corresponding to the range of estimated reverberation times. At very low frequencies, below about 20 Hz in this room, the concept of reverberation time itself becomes meaningless.

Overall, the results confirm that the initial assumptions of Section 2.1 of the way in which structural vibrations are converted to airborne sound energy in a room were reasonable and that, uncorrected, they are capable of giving results to within 16 dB over the whole measured frequency spectrum. The practical corrections for a typical drama studio have been measured and can be applied to further improve the accuracy of the predictions.

3.5 Attenuation of vibration as a function of distance

At the outset of this work it had not been decided where the most sensitive areas were to be located on the Langham Site. It was therefore necessary to attempt to measure the rate of attenuation of the ground-borne vibrations as functions of the distance from the vibration source and of the frequency. Six accelerometer points were set out in an approximately straight line at the sub-basement level in the existing building. The first of these was as close to the line of the train tunnel as possible and was used as a reference position. For each of the five remaining positions vibration levels relative to those at the reference position were recorded as functions of frequency during the passage of trains. The distance of the furthest position from the reference was 77 m. Again, only the vertical translation components of the vibration levels were recorded (see Section 3.2.2).

The results were extremely variable and are not given in their original detail here. Although care was taken to try to establish the details of the substructure and to try to avoid local structural resonances it was not possible to avoid all of the variations. The measured results therefore include substantial contributions from these unwanted factors. In an effort to reduce these effects, the data obtained has been processed to reduce some of the variations by averaging. Considering the whole set of results as a rectangular array with axes of frequency and distance, first a linear, running average over five 1/3rd octave frequency bands was applied to the data in the direction of the frequency axis. At the cost of a reduction in the frequency resolution, this reduced the variations by simple arithmetical averaging. Secondly, on the assumption that some law of attenuation which is linear in terms of dB per doubling of distance is applicable, the distances were expressed in those terms and a linear regression applied to the five data values for each frequency band. By this means, an averaging process was also applied to the data array in the direction of the distance axis.

The result of these two averaging processes is a single figure for the rate of attenuation in terms of dB per doubling of distance at each frequency. Fig. 28(a) shows the results obtained for the first measurement. Because the data on which these results are based was very irregular, a second set of measurements was carried out at different measurement points to check the validity of the results. This second set, processed in the same way, produced the results shown in Fig. 28(b). It is clear that the choice of measurement positions was not a significant parameter.

It should be remembered that the results shown in Fig. 28 are obtained after averaging over five frequency bands. They have, therefore, been convolved with a rectangular window of five 1/3rd octave bands width, thus reducing the frequency resolution. Even so, there are clearly different rates of attenuation at different frequencies – the frequency resolution is not so poor that the results for, say, 32 Hz are affected by or contain any information from those for 16 Hz. The overall accuracy of the results is such that an average linear regression coefficient of 0.95 was obtained for the 14 frequency bands from 25 Hz to 500 Hz. The coefficients for the four lowest frequencies were poor, between 0.30 and 0.79.

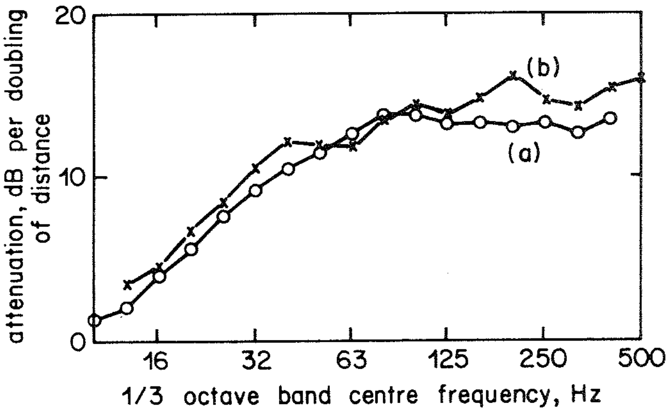

Fig. 28 – Vibration attenuation as a function of horizontal distance.
(a) Measured 14th June 1984.
(b) Measured 28th June 1984.

The results show that, at frequencies below about 10 Hz there is very little attenuation with increasing distance from the source. At frequencies above about 63–80 Hz the attenuation rate is constant at about 13 dB per doubling of distance. It is reasonable to assume that there is no sharp discontinuity between these two regimes and that the slope shown in Fig. 28 represents the true measured value of the function in this region, despite the restriction of the frequency resolution imposed by the data processing method.

These results are broadly in agreement with those obtained by others[11]. They also reflect the changes in the measured spectrum of the vibration caused by passing trains. At first, when a train is at some distance from the observer, the vibration has a predominantly low-frequency spectrum. As the distance between the train and the observer becomes less the spectrum changes to contain more higher-frequency energy. To some extent this variation is also evident subjectively in the perceived airborne noise caused by the vibrations, although it is modified by the ear's poor sensitivity at the extreme low-frequency end of the spectrum.

3.6 Attenuation of vibration as a function of height within a building

For the same reasons as in Section 3.5, it was necessary to know the rate of attenuation with height in a building. Even if the drama studios were to be located in the lower levels of the new building (which was structurally, if not acoustically, desirable) other types of studios, of which there were intended to be many, could also need to be protected against vibration from the trains. Also, if a significant reduction in vibration levels with increasing height were to be obtained then there would be an advantage in placing the most sensitive areas, such as the drama studios, higher in the building. The positioning of these large and probably heavy areas would have a very profound effect on the building design, especially if they were to be located at a significant height above the basement level.

It was not, of course, possible to measure the vibrations in the proposed building so a structure was selected which was similar to that which might be used for the proposed building and which was sufficiently close to the train tunnels to permit meaningful measurements. This was the West side of the existing Broadcasting House extension.

It was intended that the measurements should compare the train-induced vibrations at the various levels within the building with those existing at the lowest accessible level, with as few additional variables as possible. The measurement sites had therefore to be structurally similar on the different floors, with good access routes for cables and to be as close to the train line as possible. Each measurement point had to be securely connected to the building structure (with no resilient covering such as carpet) and, as far as possible, free from the influence of other building elements, such as floor slabs. The points chosen eventually were near to the stairwell of Staircase 7, exactly above each other on the different floors and, as far as could be determined, structurally identical. The lowest accessible point was in the sub-basement plant room and close to the lowest limit of the building excavation. The sub-structure to this point could not be determined but it is likely to be the foundation slab for the entire building.

The measurements were carried out in the same way as for those of Section 3.5, the lowest measurement point being used as a reference and the differences in vibration levels during the passage of a train between that reference and the remaining points being recorded, in 1/3rd octave bands. Measurements were made from the basement level to the eighth floor, a total of 11 floors. Again, because of the limited time available only the vertical translation components of vibration were recorded.

Once again, the results were extremely variable and are not given here in detail. In all, 14 measurements were made. Of these, two pairs were repeat measurements, one was made on the eighth floor when there was no train passing and one was atypical, probably because of inadequate connection of the accelerometer to the structure. Further study of the results showed that the large amount of building services plant on the ninth and the tenth floors of the building was causing more vibration than the passing trains, for all floors at frequencies of 25 Hz and 160 Hz and for all floors above the fourth at other frequencies. Again considering the data as a rectangular array, consolidation of the data consisted of replacing the data rows for the two frequencies particularly affected by the plant vibration by the average of the two adjacent rows, replacing the anomalous data column by the average of the two adjacent column and combining the repeat measurements into single columns. The variability of the remaining data was then reduced by applying a two-dimensional, weighted running average to the data array. This weighted average consisted of replacing each data value in the array by the average of the 3 × 3 array of adjacent points. This was done twice so that the end result was equivalent to a 5 × 5 window with an approximately triangular, two-dimensional weighting function. The final result of this data processing is shown in Table 1 which shows the vibration levels, relative to the reference position at the sub-basement level, as functions of frequency and height.

Table 1: Vibration levels as a function of height and frequency

Frequency	FLOOR LEVEL						
Hz	B	LG	G	1	2	3	4
10	−2	−2	−2	−2	−1	0	0
12.5	−2	−2	−2	−2	−1	0	0
16	−2	−2	−1	−1	−1	0	0
20	−1	−1	−1	0	0	1	1
25	−1	−1	−1	0	0	1	1
32	−1	−1	−1	0	0	0	0
40	−1	−1	−1	−1	−2	−2	−2
50	−2	−2	−2	−2	−2	−3	−3
63	−1	−1	−2	−2	−3	−3	−4
80	−1	−1	−1	−2	−3	−4	−5
100	−1	−2	−2	−3	−4	−5	−6
125	−3	−3	−3	−3	−4	−5	−6
160	−4	−4	−3	−3	−4	−5	−6
200	−4	−4	−4	−4	−5	−6	−7
250	−4	−4	−4	−5	−6	−8	−9
315	−4	−5	−5	−6	−8	−10	−11
400	−5	−5	−6	−7	−9	−11	−13
500	−5	−5	−6	−7	−9	−12	−13

Two principal features are evident in Table 1. Firstly, at frequencies below about 50 Hz there is very little attenuation with increasing height. Secondly, there is very little difference between those floors which are at or below the ground. Only at high frequencies and those floor levels which are at or above the level of the ground is there any significant attenuation with increasing height in this building. Again assuming that there is some linear relationship, this time in terms of dB attenuation per floor, a linear regression can be applied to the data of Table 1, for those floors above ground level only, to obtain the attenuation rate for each frequency. Fig. 29 shows the results obtained. The coefficients of regression were all in the range 0.92–0.99 with an average value of 0.97. They are, therefore, fairly reliable. It is clear that there is very little attenuation with height in this type of building, even at 400–500 Hz only about 2 dB per floor can be expected. In the frequency range most critical for underground train vibration, around 50 Hz, there is essentially zero attenuation with increasing height. At very low frequencies there is evidence for some amplification of the vibration levels at the higher floor levels (probably as a result of many, superimposed vibrational modes). Because of the averaging process used in the data processing the frequency resolution of both Table 1 and Fig. 29 is less than that implied by the number of data points. However, it is clear that these effects are not strong functions of frequency and that the resolution is adequate to describe the general trends.

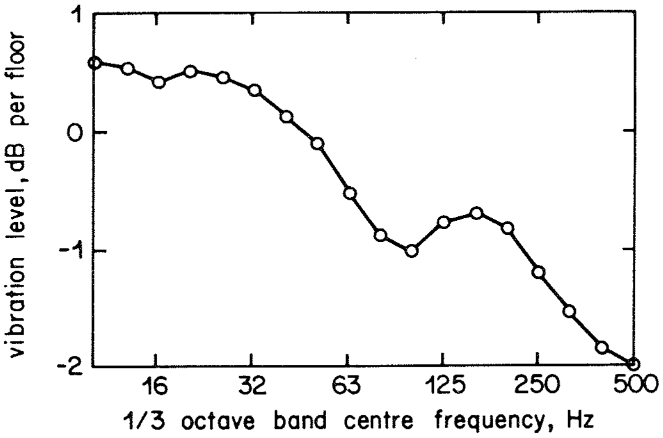

Fig. 29 – Vibration attenuation as a function of height.

It is arguable as to whether this building is typical of modern types of building construction and that the existing stairwell in particular is much more massive than a modern building would be, thereby conducting vibration more effectively. However, if a new building were to be designed with the drama studios well above ground level, to take advantage of the improved attenuation of the 'lightweight' structure, then, in order to support the inevitably great weight of the drama studios at such a height, the building would need to be of massive construction!

3.7 Measurements at Maida Vale, Studio 7

As an early experiment in the investigation of vibration isolation of large structures a full-size drama studio was constructed on a set of industrial 'vibration isolators'. This studio was subsequently to become Maida Vale Studio 7. The studio weighs about 450 tonnes and is supported by 98 large helical steel springs, each with a maximum load rating of about 55kN. The design static deflection was 22 mm, which was also achieved in practice. This, theoretically, should result in a fundamental resonant frequency of 3.36 Hz. The springs were supported on dwarf walls giving an effective airspace height of 1 m. One of the reasons for carrying out this experiment at Maida Vale was the comparative absence of ground-borne vibrations at the site. If, for any reason, the isolation system did not work as expected the the future use of the studio would not be jeopardised. It was not actually necessary to provide isolation from external ground-borne vibrations at Maida Vale, even for a drama studio. However, some vibration isolation was required to prevent interference from the nearby music studios, some of which regularly contained very high sound levels. It is well recognised, see for example References 12 and 13, that the airborne noise protection system in such areas should include attempts at vibration isolation of the structures; however, this would have been on a rather smaller scale than the experimental arrangement.

Great care was taken in the construction of this experiment to avoid any weaknesses such as might ordinarily occur in building work, such as builder's rubble being swept away down into the wall cavities, and which had given rise to doubts about the standards of construction of other attempts at vibration isolation in the past.

The low levels of site vibration which were the reason for the choice of the location for this experiment were, by their smallness, themselves the source of further difficulties. In order to measure the effective vibration isolation of such a system it is necessary to measure, in some way, the difference between the vibration levels on the two sides of the isolation system. If there is no vibration then the isolation cannot be measured; less extremely, the isolation can only be measured up to the limit set by the difference between the source vibration levels and the inherent noise level of the measuring equipment. With conventional, commercially obtainable accelerometers at Maida Vale the measurement limit was very small. Only two alternatives were available – either to develop more sensitive accelerometers or to generate additional ground vibrations.

A pair of commercially available electromagnetic transducers which were intended for use as shakers were modified for use as accelerometers by supporting them 'upside down' with extra suspension systems in frames. The mass of the magnet assembly became the inertial mass and the power winding became the sense coil. The vibration input was via the output strut. The arrangement was very sensitive, it would light a small electric light bulb from the vibrations generated by footsteps, and had an output response proportional to the velocity of the vibration. It was, however, rather delicate and had to be clamped before being moved. It also had one fundamental problem which became evident later, after most of the measurements using it had been made. This problem was Newtonian friction in the internal suspension. At the very small amplitudes of vibration existing naturally at the site, at most about 70 nm rms (1/10th of the wavelength of red light), the friction component is significant because it appears as a 'short circuit' across the main suspension, thus altering the frequency response of the transducer. When used for its intended purpose as a shaker this friction is not a problem because the movements involved are always much larger and the friction component reaches a limiting value which is small compared with the other forces.

Small variations in the levelling of the transducer made significant changes to the friction component. The transducers had been developed in the naturally smooth and level conditions of a laboratory but on the rough concrete surfaces of the Maida Vale site slab it was impossible to obtain consistent levelling. Therefore most of the results obtained using these transducers at Maida Vale were useless and the remainder were questionable. However, four measurements were thought to be sufficiently reliable to be included in this report. Fig. 30 shows the average of the measured isolations obtained, in 1/3rd octave bands, together with the theoretical isolation predicted by the simple, three-element model. Although the result is rather irregular, some features are evident. The fundamental resonance at about 5–6 Hz is fairly clearly defined. It is also clear that the theoretical values of isolation are not obtained at frequencies well above the fundamental resonance. At frequencies above 25 Hz the temporal correlation between the vibrations on the two sides of the isolation system had disappeared although the transducers were still measuring real vibrations. It can be concluded that there were other, unrelated sources causing vibration in the mass of the studio. At the time of these measurements, which pre-dated those on the Langham Test Slab, it was thought that the measurement technique was responsible for the poor results and, in any case, because of the low vibration levels and the consequential problem with extraneous vibration, the upper frequency limit was unacceptably low. The method was not pursued further.

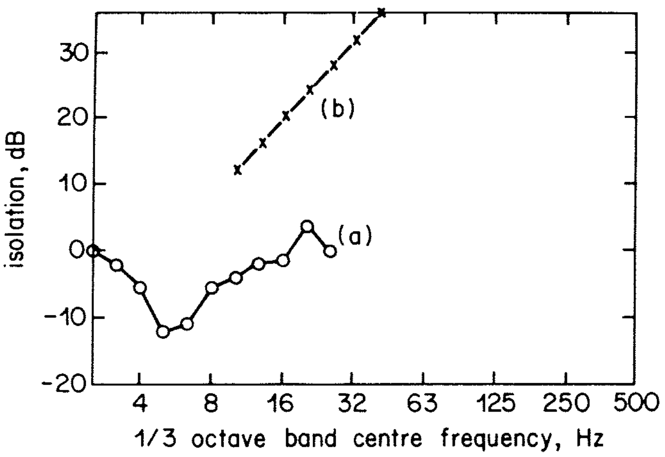

Fig. 30 – Vibration isolation at Maida Vale, Studio 7, measured using natural site vibrations.
(a) Measured values.
(b) Theoretical isolation of simple model.

The alternative method using conventional accelerometers and high levels of vibration generated artificially was next investigated. The generation of significant levels of vibrations in the foundations of such a massive construction is no easy matter. The forces required are very large and, to produce a vibration distribution pattern representative of a relatively distant source requires either very many synchronised sources or a single source at a distance. However, the option of a single powerful source at a distance was effectively ruled out because of disturbance to the adjacent, domestic properties. Some consideration was given to vibration generators involving vibrating or rotating eccentric masses but a few simple calculations showing potentially dangerous quantities of stored kinetic energy made this approach less than attractive. Also, the vibration sources had to be manipulated in the 1 m high airspace beneath the studio and had, therefore, to be reasonably portable. Eventually, a method using explosive charges was developed. A small charge, restrained vertically by a superimposed large inertial mass, provided an impulse of force to the surface beneath. The mass was in the form of damp sand in manageably small bags which could be replaced on top of the charge after each firing. A second function of the sand was to reduce the acoustic noise produced by the explosion which was otherwise very high. Using this method it was not possible to synchronise other devices so that the distribution of the induced vibration pattern was not ideal. Fig. 31 shows the average of four measurements of vibration isolation made in this way, again together with the theoretical isolation predicted by the simple, three-element model. The results show very little

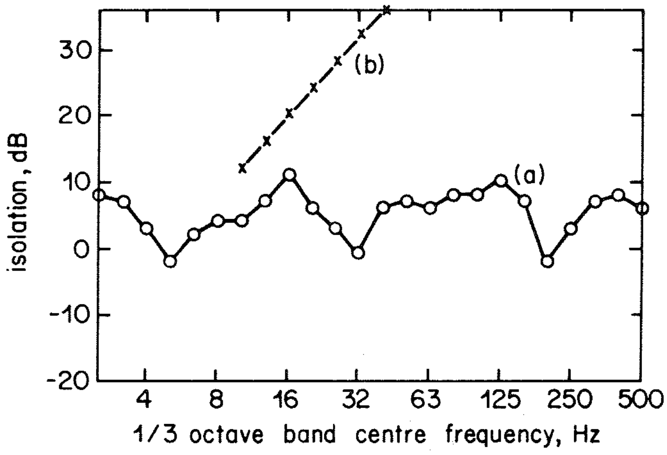

Fig. 31 – Vibration isolation at Maida Vale, Studio 7, measured using explosive excitation.
(a) Measured values.
(b) Theoretical isolation of simple model.

measured isolation but, at least up to a frequency of 25 Hz, they are very similar to those obtained using the natural site vibrations, shown in Fig. 30. It became clear, following the later work carried out on the Langham Test Slab, that the acoustic noise, in this case generated by the explosion, was a limiting factor. The method is only usable where there is additional airborne sound insulation between the site of the explosion and that of the vibration measurements.

This early work at Maida Vale mostly served to show that the measurement of vibration isolation in large structures is difficult, especially if there is no usable natural site vibration. Following the Langham work and the identification of the acoustic flanking path as a major component, it is now thought that the measurement results shown in Figs. 30 and 31 are true measurements, admittedly with large variations due to errors, of the attenuation of the isolation system. This indicates that the apparent failures of earlier attempts at vibration isolation may not have been as a result of poor construction.

One aspect which was illustrated clearly by this experiment was an anomaly in the fundamental resonant frequency. The measurement of the static deflection of the springs had been carried out carefully by the structural engineer and was close to 22 mm. This should have given a resonant frequency of 3.36 Hz (± 0.15 Hz) as there is no substantial non-linearity mechanism in this type of spring. However, the actual frequency was measured to be about 4.75 Hz (± 0.25 Hz). The origin of this discrepancy is probably the extra stiffness of the closely enclosing airspaces, bounded by the surrounding building structures both above and below the studio.

It is not directly an acoustic matter but the experiment also served to show how such rooms could be constructed. The structural engineers considered many possible methods of achieving the desired result – that of a heavy and relatively fragile masonry shell balanced on a set of springs with more static deflection than the structure could accomodate differentially without cracking disastrously. The method finally chosen was to build the shell on props at the desired final height. Springs with built-in screw jacks could then be inserted in the spaces between the props and the jacks screwed up progressively and evenly until the springs had been compressed by the calculated amount. If the calculations were correct then the springs would then be taking all of the weight and the props could be removed. As far as could be determined, the method worked perfectly except that allowance had to be made for the compressibility of the temporary props.

3.8 Measurements in Studio B11.

Studio B11 in the Broadcasting House extension is another example of a large drama studio supported by a vibration isolation system. It was constructed in the early-1960's and uses a layer of rubber compression mat over the whole of the floor area. It was possible to inspect the edge of the mat through a cable-duct access hole. It is the usual ribbed type of mat but with relatively deep ribs. It appeared to be formed of unusually soft rubber but did not appear to be degraded significantly. The total thickness of the mat is about 25–30 mm and the static deflection appeared to be about 3 mm. It is thought that the remainder of the studio was built on the edge of the floor slab so that the rubber mat supports the entire weight of the studio. This studio is one of those which had long been thought to have a defective vibration isolation system. Certainly its subjective performance falls far short of the requirement: the underground trains are clearly audible in the studio and cause serious disturbance to its users.

Five measurements of the vibration isolation of this structure were carried out, access to the supporting floor slab being obtained from the room below. The results are shown in Fig. 32(a) for 1/3rd octave band analysis. It appears from these results that the fundamental resonant frequency is about 25 Hz, implying a static deflection of 0.4 mm, if the stiffness of the rubber mat remains reasonably constant. This is very different from the observed apparent deflection, so much so that one or other must be incorrect. It is likely that the conditions at the access hole were not typical of the floor as a whole, but it is also possible that the actual static deflection is very small, though probably not intentionally as small as the

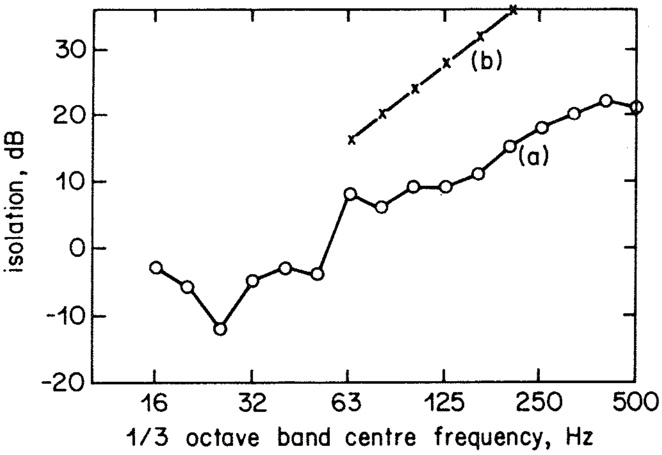

Fig. 32 – Vibration isolation at Broadcasting House, Studio B11, measured using train excitation.
(a) Measured.
(b) Theoretical performance of a simple, ideal system with a resonant frequency of 25 Hz.

results indicate. Given such a high resonant frequency, significant vibration isolation at low audio frequencies would not be obtained even from a perfect system. Fig. 32(b) shows the theoretical performance of the simple ideal system with the same fundamental resonant frequency, for frequencies well away from the resonance. It is clear that, like all of the other attempts at the vibration isolation of large structures, the performance of this one also falls short of that predicted by the simple model. However, far from being particularly defective, this one is not as far short of the ideal as was the carefully controlled Langham Test Slab. It may therefore be concluded that there is no significant defect in the construction and that this is a further example of the practical limitations of real vibration isolation systems.

Probably because of a design error, the static deflection of the system, as inferred from the resonant frequency measurement, is far too small to provide good isolation at low frequencies. Several other cases of such design errors have been found retrospectively, arising from, for example, a confusion of the distributed and the point-loading specifications for the floor.

4. FURTHER DEVELOPMENTS OF THE LANGHAM TEST SITE

4.1 Construction

Following the series of measurements on the (essentially) two-dimensional plane of the Langham Test Slab, the structure was extended by the construction of the remaining five surfaces of a closed box representing a small-scale model of a studio. The walls were constructed from dense, 150 mm concrete blocks and the ceiling from 100 mm poured concrete on permanent steel shuttering. The internal plan area was 3.6 × 4.9 m and the internal height was 2.3 m. The total mass was about 32 tonnes. The foundation slab thickness was increased to a total of about 1.2 m to accomodate this weight. A steel door, intended to have good airborne sound insulation performance, was included in one of the short walls.

Space and structural limitations made it difficult to change the mountings once the construction had been completed and it was therefore necessary to decide at the outset which kind should be used. From the results of the earlier tests on vibration isolators large helical steel springs were chosen by the Acoustic Consultant for this extension of the work. These were fitted with a different type of noise-stop pad, made of the same type of compressed glass-fibre mat as the small cubes tested earlier (Section 3.2). A comparison of the two types of noise-stop pad was made, using the completed box, but the results were not distinguishable. This was hardly surprising given that the use of such pads had already been shown to be ineffective, given the presence of the dominant acoustic flanking path and given the now much more complicated modes of the completed box structure.

Many measurements were made using this structure; of internal noise levels, of vibration isolation from the foundation slab and of the vibrational modes of the supported structure.

4.2 Structural modes

The modal behaviour of the Test Slab had been shown to be complex (Section 3.2.4). The addition of the walls and the ceiling had been expected to further increase this complexity because of the potentially much higher modal density. This was found to be the case but it was also found that the damping was much greater. These two factors made it impossible to identify individual modes with any reliability. For one mode which was marginally discernable, at 23 Hz, it was found that the two sides of a masonry joint were moving in anti-phase, thus indicating no mechanical connection between them over the greater part of the length of the joint. This was very surprising because the joint was the horizontal one at the foot of a wall where the wall was ‘supported’ by the floor slab and which would have been expected to have the weight of the wall and some of the ceiling upon it. Such uncertainties of constructional detail demonstrate the futility of attempting to predict the modal behaviour of real buildings. Frictional losses at such imperfect joints probably account for the observed higher damping factors.

4.3 Vibration isolation

The fundamental resonant frequency of the box on eight steel springs was 2.2 Hz, exactly what had been expected from the design static deflection. The average measured vibration isolation was marginally worse, by an average of 4 dB, than for the Slab alone with a similar resonant frequency, despite there being a layer of mineral wool 100 mm thick in the airspace beneath the box. This may have been because the larger surface area of the box was coupled more effectively to the surrounding air, thereby increasing the effectiveness of the acoustic coupling path.

4.4 Internal noise levels

Using the theory of Section 2 and particularly the performance limitations of Section 2.3, together with the radiation efficiency as measured in Studio B11 (Section 3.4), an *a priori* prediction was made of the maximum airborne noise levels which would occur inside the box during the passage of trains. This prediction also required values for the magnitudes of the ground vibrations and of the internal acoustic response of the box. For the former, many hundreds of samples had been obtained during the course of the earlier measurements. For the latter, an assumption about the performance of the acoustic treatment, and the consequential reverberation time, had to be made. This assumption was subsequently updated when the model studio was completed and the acoustic treatment installed.

Fig. 33(a) shows the predicted maximum internal noise level as a function of frequency, corrected for the actual, observed reverberation time. Fig. 33(b) shows the comparable measured values. It is clear that the prediction is close to the actual performance over most of the frequency range. At frequencies below 31.5 Hz the assumption made that the modal density of vibrations of the box is high becomes invalid and the prediction becomes much less accurate.

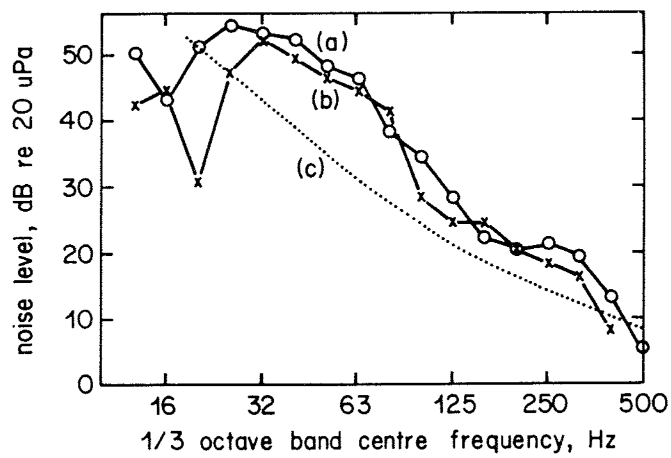

Fig. 33 – Internal acoustic noise levels in the Langham Test Box.
(a) Predicted. (b) Measured. (c) Criterion iii.

4.5 Second enclosure

Because of the dominant effect of the acoustic flanking path in limiting the maximum achievable vibration isolation it was obviously necessary to interpose some form of airborne noise attenuation. For the low frequencies at which the train vibrations caused the most interference the only effective form of insulation is masonry or some other equally heavy construction. To achieve higher values of noise isolation than had been obtained from a single enclosure it would therefore be necessary to surround the supported structure by another complete enclosure, itself on a set of noise isolators. In this way, the noise levels might be reduced by 20–25 dB by the first enclosure and a further 20–25 dB by the second enclosure, making a total of more than 40 dB. Because the vibration isolating mountings were not themselves limiting factors over the important frequency range, provided that a suitable type of vibration isolator was chosen, it would not be necessary to support the inside structure by the outside enclosure. Instead, both sets of mountings could be supported off the main foundations, but with careful sealing around the mountings for the interior box where they passed through the exterior box to avoid an airborne leakage path. This would make the design of the isolation systems simpler because there would not be the interaction which would occur if they were mechanically in series.

Such a construction, for real studios on a real site, would be structurally complex, difficult to build and enormously expensive; even as an experiment on a much smaller scale than a full-size building it was not clear how such a combination of structures could be built. In the event, because of practical limitations, an incomplete box was constructed, covering five of the six sides of the model studio. Simple arithmetic shows that at best 8 dB (10 log 6) of improvement could be expected from such an addition as a result of the sixfold reduction in the surface area exposed to the airborne sound, instead of the 20 dB minimum expected from a complete box. Fig. 34(a) shows the measured maximum internal noise levels, compared with those measured inside the single enclosure, Fig. 34(b) (Section 4.4). The measured improvement resulting from the addition of the second enclosure was only 3.7 dB, averaged over the frequency range 32 Hz to 400 Hz. It is probable that the improvement was smaller than the theoretical value because the surface which

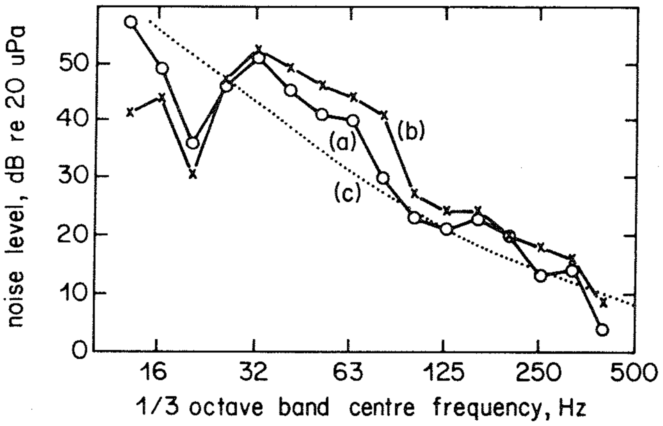

Fig. 34 – Measured internal acoustic noise levels in the Langham Test Box.
(a) Double enclosure. (b) Single enclosure
(c) Criterion iii.

was not protected, the floor, was the one which was closest to the foundation surface and therefore was better coupled to the ground vibrations than the five other surfaces. Also, the additional enclosure was not a perfect insulator of airborne sound.

Below 32 Hz, the isolation of the double box structure was up to 16 dB worse, particularly at about 14 Hz which was the fundamental resonant frequency of the outer box on its supports.

5. APPLICATION TO THE LANGHAM DEVELOPMENT

5.1 General

The results so far presented in this report have not been related numerically to the situation on the Langham site. Most have been in terms of vibration isolation and are ratios between the two sets of vibration measurements and do not depend directly on the absolute magnitudes of the vibrations.

For the derivation of numerical values for the required vibration isolation on the Langham site some factors are required in absolute terms.

The required vibration isolation is also dependent on the type and size of the area potentially subject to interference. The most critical types of areas are drama studios, of which three were intended to be built on the Langham site. All of the calculations and results in this Report which are specific to a particular area assume that the subject area is a drama studio of the dimensions and acoustic treatment of Studio B11 in the existing Broadcasting House Extension.

The background noise level in the studio from all sources is assumed to meet the current standard, Criterion iii[14].

5.2 Source vibration levels

Very many measurements were made of the source vibration levels at the site of the Test Slab during the course of the work there. These were implicit in the measurements of vibration isolation. To optimise the isolation measurement these had all been made in terms of the average level during most of the passage of individual trains. This made the measurements reasonably consistent and comparable with each other. However, the trains took more than 15 s to pass the site and the integration period was set to that value. Subjectively, the perception of interference occurs on a shorter time-scale. So, for the purposes of deriving isolation requirements, a number of special measurements of the absolute level of the ground vibrations was made on the basis of the highest level recorded with a 1 s integration time. The difference between the two bases of measurement was approximately 3 dB. At the same time, records of the unfiltered vibration signals were made on a chart-recorder over a period of about 1 week. This allowed some estimate to be made of the relative frequency of particularly 'noisy' trains.

This time-record showed three main features. The first was that the vibrations caused by the trains were, by a margin of about 30 dB, the dominant source of ground vibrations at the site. The second was that, on average, the site was affected by about one train per minute during most of the day. In the periods 09.00–10.00 hrs and 17.00–19.00 hrs the average frequency was doubled. The third feature was that a four-fold pattern was evident, reflecting the relative significances of the four tunnels close to the site, with a difference of about 6 dB between the two extremes. Surprisingly, there was very little difference in the overall amplitudes of the recorded vibrations when the trains were heavily loaded, compared with when they were only lightly loaded. However, the spectrum of the vibrations did shift towards the lower frequencies when the trains were heavily loaded.

It could therefore be concluded that one train in every four would consistently cause vibration levels close to the maximum recorded levels and that this potential interference would occur on average every two minutes during the peak periods and every four minutes at other times. Between these peaks, three other trains would cause vibrations within 6 dB of the peak levels. With such a pattern of potential interference there was no alternative but to design the vibration-isolation system for the worst-case.

The spectra of 17 trains, in 1/3rd octave bands and with an integration time of 1 s, were analysed in detail. Inevitably, the spectra showed variations as a result of the individual train parameters, such as speed, load and the state of the tyres on the wheels. To obtain a representative design spectrum the highest levels of vibration for each frequency band were combined into a composite spectrum which therefore represents the highest level of vibration which is likely to be encountered at the measuring point for all trains. No individual train would be expected to generate those levels in all frequency bands simultaneously. Fig. 35(a) shows the resulting spectrum. This was the assumed source vibration levels for all of the numerical calculations in this Report, including the predictions of internal noise levels in Section 4.

5.3 Permitted surface vibrations for a drama studio

In Section 2.1 the theoretical relationship between the vibrations of the boundary surfaces and the resulting internal noise levels within a room were derived, given some initial assumptions. In Section 3.4 the results are given of measurements in a typical, existing drama studio verifying the theory and giving correction factors to account for the errors in the assumptions at the extremes of the frequency range (Fig. 27). From these two sets of data the maximum permitted vibration levels in the boundary surfaces of a drama studio can be calculated. The assumption is made that the noise generated by the trains will not be perceptible if it just reaches the same level in each 1/3rd octave frequency band as the existing, continuous background noise from the studio ventilation plant (this is the same assumption as is made for all sound insulation work, including the more usual airborne sound insulation and is based on the audibility of wide-spectrum sounds in combination). Fig. 35(b) shows the result of the calculation.

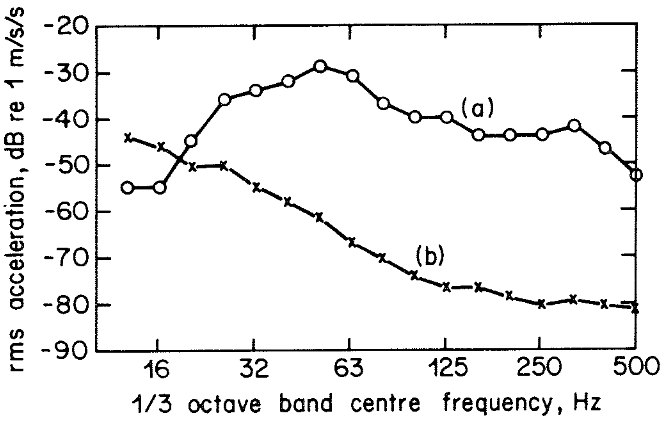

Fig. 35 – Vibration levels for Drama studios on the Langham site.
(a) Source spectra.
(b) Maximum permitted surface vibrations

5.4 Vibration isolation requirements

The vibration isolation required by a drama studio as a function of frequency is simply the difference between the two sets of numbers illustrated by Fig. 35. However, the measured spectrum of source vibration levels is for the particular point at which those measurements were carried out. Allowance has to be made for the exact location of each studio. The measurement point was located as close to the line of the train tunnels as was possible on the site. It was approximately 18 m laterally and 20 m vertically from the centre-line of the tunnel. The final position of the three drama studios on the site had not been decided but was tentatively in a line roughly parallel to the train tunnels at around 40 m laterally and at about the same horizontal level as the highest of the tunnels. Thus the typical distance of the proposed studios was not very different from that of the measuring point, although at a different angle. Thus no allowance could reasonably be made for any attenuation with distance or with height, despite the work and the results described in Sections 3.5 and 3.6. In any case, the results for the attenuation with height showed that, for one building at least, there was no significant difference between any of the floors which are below ground level.

Fig. 36(a) shows the requirement for the vibration isolation of the drama studios at their tentative locations. It is numerically equal to the difference between the two lines on Fig. 35.

All of the above assumes that the drama studios are to be built directly on the foundations of the new building, in the same way as the Test Slab was on those of the old. If they are not and there is an intervening floor structure then the resonances of this intermediate structure will amplify the source vibrations at some frequencies. It is difficult to predict the magnitude or the frequency of such resonances, and the figures given here, in Fig. 36(a) for vibration isolation must be considered to be the nett effective isolation, including the effects of any amplification by the supporting structure. Such resonances increase the demands made of the main vibration isolation system, to an extent equal to the amplification by the intermediate structure.

5.5 Comparison of the target with achieved isolations

Fig. 36(b) shows the idealised vibration isolation performance of a simple system with a fundamental resonant frequency of 3.2 Hz (the same system as described in Section 2.3.5 and the results as given in Fig. 9(a)). When compared with Fig. 36(a) it

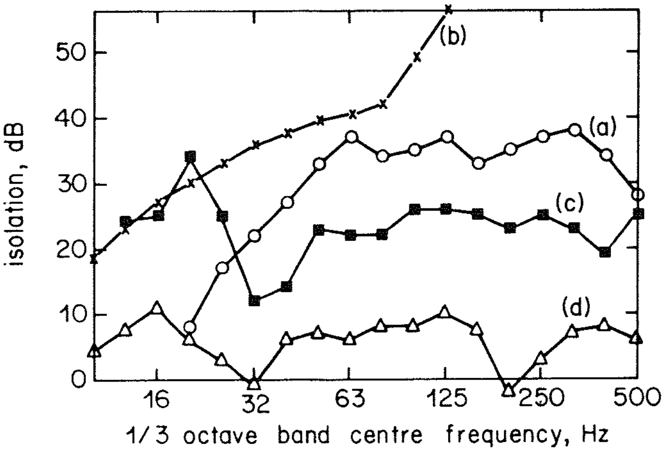

Fig. 36 – Vibration isolations for Drama studios on the Langham site.
(a) Required isolation to achieve Criterion iii.
(b) Theoretical isolation for a fundamental resonant frequency of 3.2 Hz.
(c) Best achieved result using the Test Slab.
(d) Results from Maida Vale, Studio 7.

shows that, ideally, such a system could meet the requirement. However, Fig. 36(c) shows the best measured performance obtained using the Test Slab and Fig. 36(d) the measured performance of the vibration isolation system installed at Maida Vale, Studio 7. It is equally clear that practical simple systems fall far short of the requirement based on Criterion iii.

5.6 Direct perception of vibration

In addition to generating acoustic noise which may be above the permitted limit in an area, surface vibrations may be directly perceptible to the occupants through the senses of touch or of balance. Reference 15 gives some figures for the sensitivity of the human body to whole-body vibrations. The values given there for the boundary between 'imperceptible' and 'just perceptible' vibrations were converted to the same metric units of acceleration used in the remainder of this work. These values were then compared with the measured source vibration levels. Over the frequency range from 8–500 Hz (in octave bands) the actual source vibrations were less than the threshold of perception. The least margins were in the 31.5 and the 63 Hz bands where they were 13 and 16 dB respectively. At other frequencies the margins were between 28 and 56 dB. Thus, over the frequency range for which the vibrations caused by Underground trains are significant, there is a reasonable margin of safety and the direct, tactile perception of the train vibrations is unlikely whatever the construction of the building and the studios.

There is however a second aspect to the problem of the direct perception of vibrations in vibration-isolated building structures. At the frequency of the fundamental resonance of the vibration isolation system the room is very susceptible to vibration excitation because of the amplification effect at resonance. The magnitude of the effect is dependent on the 'Q'-factor of the system and may be very large indeed. Even if there is no external source of vibration at or near to the resonant frequency the occupants of the room will generate some excitation by the their normal movements. This was clearly demonstrated by the work done on the Langham Test Slab. It is self-evident that heavier structures are less susceptible, because they are less influenced by a particular level of excitation. It is also apparent from the literature, for example References 15 and 16, that high frequencies are less perceptible than low (at least over the frequency range relevant to this work). For the Langham Test Slab with its enclosure, a mass of about 32 tonnes and a resonant frequency of 2.2 Hz, the effect was very significant. All but the most careful movements by the occupants caused perceptible oscillations which, with less care, bordered on the nauseous. For Maida Vale, Studio 7, with a mass of about 450 tonnes and a measured resonant frequency of 4.75 Hz, the effect is completely negligible.

It is difficult to be precise about the subjective limits of this effect because there is no data available beyond that which has been obtained during the course of this work. It is different to the widely-covered susceptibility of humans to whole-body vibration because it consists of vibration which is induced by the room's occupants themselves. Establishing limits by means of subjective tests would require the construction of a range of rooms of different masses (up to several hundred tonnes) and with different resonant frequencies – an obviously difficult and expensive undertaking. The experience of Maida Vale 7 indicates that a full-size drama studio is likely to be acceptable if its fundamental resonant frequency is about 3 Hz or more, but that significantly lighter structures than that may be marginal.

6. PROPOSED CONSTRUCTIONS

It is clear from the results shown in Fig. 36 that the vibration isolation requirements of a drama studio meeting the normal acoustic standards could not be satisfied on the Langham site by a simple, single stage of isolation. It is also clear that, within the normal audio frequency range and for large structures, the airborne flanking path is the main limitation. Means must therefore be provided to reduce the significance of this path by some form of sound insulating barrier which is effective at low frequencies. In practice, only heavy masonry constructions are capable of providing such insulation within a building. The degree of sound insulation required is about 20 dB at frequencies down to 25 Hz, implying at least the equivalent of 225 mm thickness of concrete[17]. This airborne noise barrier must itself be insulated from the ground vibrations by a vibration isolation system, otherwise it becomes part of the ground structure and thus serves no vibration isolation function. All of this shows that some form of double vibration isolation system is necessary.

Such a double isolation system can take many possible forms, ranging from a relatively close-fitting shell around the individual studios to the isolation of the entire building as the first stage of isolation. Each of these variations has its own set of advantages and disadvantages. The close-fitting shell can be relatively lightweight, as it only has to support its own weight and provide the necessary airborne sound insulation, but it would be difficult to build within the small access spaces and very inefficient in the use of the space within the building. Also, as outlined in Section 4.5, the detail of the vibration isolating mountings for the inner shell would be difficult. They would either have to pass through the structure of the outer shell, with effective seals to prevent the transmission of airborne sound, or be supported by the outer shell, in which case the interactions between the two sets of mountings would cause further design problems because of the similarity in the masses of the two supported structures.

Supporting the entire building on vibration isolation mountings also has problems. Structurally, the design of the building and its foundations would be complicated by the need to concentrate the loadings at the mountings. However, most recent buildings in London are supported on a relatively small number of piles so the same problem has had to be solved for those cases anyway. Also, the dynamic loadings on the building, particularly wind pressure, cause design difficulties in providing sufficient stiffness to resist lateral forces without compromising the vibration isolation performance. Nevertheless, there are very many large buildings throughout the world which are supported on vibration isolation mountings, including the London 'Post Office Tower'.

Significant advantages arise from the isolation of the entire building as the first stage in the isolation of individual studios. First, the mass of the building is large in comparison with that of the studio; this means that there is little interaction between the two sets of mountings and their individual isolation characteristics are additive. Second, the space within the building is used more effectively because there is only one isolation system for each studio there; effectively the spaces occupied by the individual second stages of isolation have been concatenated and put on the outside of the building. Third, the large scale of the mountings required to support many thousands of tonnes means that it is relatively simpler to obtain the large static deflections required for low-frequency vibration isolation. Fourth, because the isolations of the two systems are additive, each system becomes less critical; in particular, the low-frequency limit of the individual systems for the studios can be higher. This means that smaller static deflections and somewhat simpler and cheaper constructions can be employed in the second stage.

A third strategy is a compromise between the two extremes described above. In this the majority of the building, containing all of the non-sensitive areas such as offices, is constructed in the usual way. Within that structure, the sensitive areas are grouped together in a sub-building which has vibration isolation systems separating it from the ground and from the remainder of the building. The individual sensitive areas would then have their own vibration isolation systems. Using such an intermediate design, all of the advantages of the extremes are retained without any of the disadvantages. For the Langham site, with three drama studios and ancillary areas and some other types of sensitive studios, the sub-building would be heavy enough to permit reasonably independent behaviour of the two sets of isolation systems and would be protected from the external forces by the surrounding conventional building. One problem which would need to be solved would be the bridging of the fixed part of the building over the top of the isolated part of the building. A simple approach to this problem would lead to very long spans and consequently very deep trusses, which would again be wasteful of space inside the building.

Using the methods and the data already employed for the successful prediction of noise levels in other structures (Section 4.4), Fig. 37(a) shows the estimated performance of such a compromise design. The assumptions made were that the static deflections of the main building mountings were equivalent to a resonant frequency of 3.2 Hz, that is, 25 mm with no non-linearities, or about 40–60 mm for typical rubber compression mountings, and those of the studio mountings were equivalent to a resonant frequency of 7.5 Hz, or about 4–5 mm static deflection. The masses of the studio and the sub-building were assumed to be 450 and 4500 tonnes respectively. However, if the ratio of the

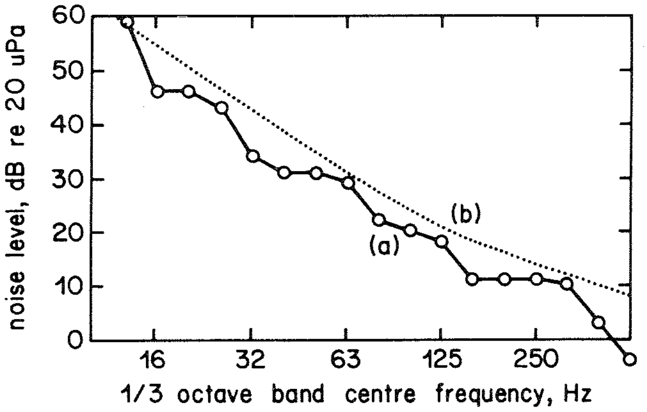

Fig. 37 – Predicted performance of the proposed double isolation system.
(a) Estimated. (b) Criterion iii.

masses is of that order, the actual ratio is not significant.

Fig. 37(b) shows again the normal background noise criterion for drama studios. It would appear, from a comparison of Figs. 37 (a) and (b), that this proposed construction would just satisfy that requirement.

7. AN ALTERNATIVE PROPOSAL FOR THE ISOLATION OF THE DRAMA STUDIOS

The constructions discussed in Section 6 would undoubtedly be expensive to implement, financially and in terms of space on the site. The senior management of BBC Radio and Engineering took these factors into account, along with advice from the BBC Acoustics Committee that noise Criterion iii should be observed as usual, when considering a proposal by the Acoustic Consultant that a simple, single stage isolation system should be used, provided that a less stringent noise criterion would be acceptable. Also, they listened to relevant simulations of the Underground train noise. On balance, they decided that the non-acoustic factors were so important that the less stringent criterion was acceptable. Accordingly, the criterion for that noise in the new drama studios on the Langham site was set to be equal to the levels the Consultant predicted could be obtained using a simple, single stage isolation system. This criterion is shown in Fig. 38(a), compared with the usual criterion for drama studios (iii), shown in Fig. 38(b).

The essential features of the Consultant's proposal for the vibration isolation of drama studios on the Langham site included a floor and walls which were separately isolated on different sets of helical steel vibration isolating mountings. The design static deflection of the floor was 25 mm, giving a theoretical fundamental resonant frequency of about 3.2 Hz. The walls and ceiling were to be supported on mountings with the very large static deflection of 150 mm. The proposal did not include any sound insulating barrier between the ground and the studio floor or between the studio ceiling and the structure above, although it did include a second set of separately isolated walls.

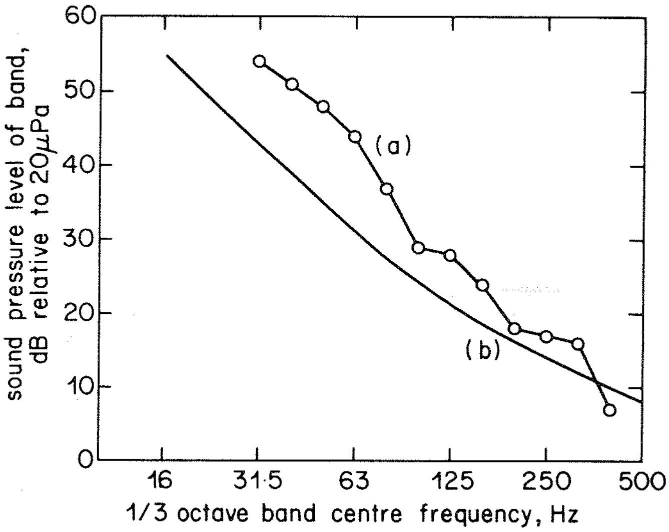

Fig. 38 – Drama studio background noise criteria.
(a) Criterion for drama studios on Langham site proposed by Acoustic Consultant.
(b) Normal drama studio Criterion iii.

One of the reasons for the structural details of the Consultant's proposals was that the structural engineers foresaw the possibility of 'ground heave' as a result of the removal from the ground of the weight of the existing building. This was expected to amount to a maximum of 25 mm over a distance equal to the length of a drama studio. Therefore, any design had to be capable of accommodating that amount of vertical distortion. The Consultant's proposal achieved this by having a floor-slab which was flexible enough and sufficiently frequently supported to bend and thus conform with any changes in the foundations. The supports for the walls and the ceiling were intended to be sufficiently compliant that a change in their loading corresponding to a change of 25 mm in static deflection would be acceptable. This solution avoided the need for large spans in both the floor and the wall structures.

Fig. 39(a) shows the Consultant's prediction of the performance of this structure in isolating the ground vibrations, which is identical to the target background noise criterion shown in Fig. 38(a).

Fig. 39(b) shows the authors' estimated performance of this proposal, using the methods and the data already employed for the prediction of noise levels in other structures (Section 4.4). For this

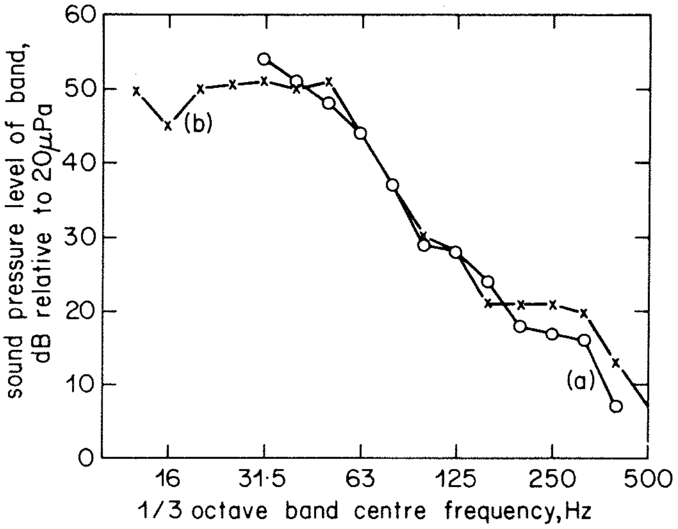

Fig. 39 – Predicted noise levels resulting from Consultant's proposal.
(a) Consultant's prediction.
(b) Authors' estimate.

prediction it was assumed that the walls and the ceiling were perfectly isolated and that only the floor contributed to the studio noise level.

There are, however, also the potential problems of how such a construction could be built. There would be difficulties in constructing the walls and ceiling of a full-size drama studio on a set of mountings with a static deflection of 150 mm and in designing the wall to be strong enough to bridge over the potentially somewhat indeterminate static deflections.

But, the most serious difficulties would probably be those of the movement of the floor and the ratio between 'live' and 'dead' loads. One of the fundamental problems, which was clearly demonstrated by the work done on the Langham Test Slab, was the perceptibility of vibrations, at the fundamental resonant frequency, induced by normal room occupancy, as described in Section 5.6. For a typical size of drama studio using the Consultant's proposed construction, the floor structure would have been unlikely to exceed 50 tonnes and had an intended resonant frequency of 3.2. Hz. It is likely that its behaviour in this respect would be closer to that of the Langham Test Slab (34 tonnes/2.2 Hz) than to Maida Vale 7 (450 tonnes/5.5 Hz). Also, the design rated normal occupancy loadings, of about 1–2 kPa, would be a significant fraction (perhaps up to one third) of the loading on the mountings. If the floor were loaded unevenly then it could be caused to tilt noticeably.

The design of the mountings and the detail of the joints between the floor and wall structures would need to accommodate these movements, together with those resulting from ground movements. The design and sealing of the joints between floor and walls to give adequate airborne sound insulation whilst permitting possible movements on the necessary scale would be difficult.

8. CONCLUSIONS

Many existing areas in Broadcasting House are subject to severe ground-borne vibration from the trains of the London Underground. Mostly because of the proposed development of a new Broadcasting Centre on the nearby Langham site, a large amount of investigation into the problems of vibration isolation has been carried out, particularly of large structures. This has taken place over a period of about eight years and has included theoretical studies of the behaviour of vibration isolation systems, measurements in existing studios and the development of a test facility in the basement of the existing building on the Langham site.

As a result of this work, the technical reasons for the apparent failures of previous attempts and the effects of two known fundamental limitations of such isolation systems have been confirmed. Limitations of some types of commercial vibration isolation mountings as a result of unwanted modal vibration responses have been demonstrated. For the isolation of large structures, additional limitations due to the modal vibrations of the structures themselves and, most importantly, the severely limiting effects of the acoustic coupling have been identified. The acoustic coupling effect was found to be much larger than had been suspected and limited the achievable isolation of a studio-like structure to a maximum of about 25 dB at all frequencies, even under ideal experimental conditions. This factor alone is sufficient to account for the apparent failure of the previous attempts at vibration isolation in Broadcasting House and other places.

Measurements carried out on the test structure and in existing areas have enabled predictions of the performance of proposed isolation systems to be made with resonable accuracy. These prediction methods have shown that single vibration isolation systems are unlikely to result in studio background noise levels which satisfy the usual BBC criterion for a drama studio and that a double isolation system, incorporating airborne sound insulation, would be necessary to meet that criterion for the drama studio background noise level on the Langham site.

The difficulties and expense of constructing a double isolation structure are considerable but a proposal which minimises these whilst retaining the degree of isolation necessary to meet the normal criterion has been outlined.

This proposal requires that a significant portion, but not the majority of the whole building be supported as a single unit on a set of primary vibration isolating mountings. Each studio and other sensitive area would then be supported within this sub-building on a further set of mountings. However, this second set of mountings would require a much lower performance than if they were the primary isolation and would thus be significantly cheaper and easier to install.

The methods used in this Report for the calculation of the performance of a vibration isolation system are somewhat empirical. More rigorous methods, such as finite-element analysis, could be employed, assuming that completely detailed construction drawings were available. However, all building work is ultimately subject to the uncertainties of actual construction. Such methods might indicate more accurately the best which could be achieved – in practice, they may be no more exact.

In the case of the Langham site itself, non-acoustic constraints required Radio and Engineering managements to relax the usual acoustic criterion and thus the Acoustic Consultant was able to propose an acceptable system based on a simple, single stage of vibration isolation. The evidence from the Test programme, however, indicated that even if it had proved to be acoustically and structurally adequate, some problems may have been encountered by the subsequent users of the studios because of the instability of the structure.

Although much of the work, especially that on the Test facility, was carried out under the direction of the Acoustic Consultant, the views and interpretations expressed in this Report are those of Research Department.

9. REFERENCES

1. SNOWDEN, J.C. 1979. Vibration isolation: Use and characterization. *J. Acoust. Soc. Am.* **66,** 5, November, 1979.
2. BERANEK, L.L. 1954. Acoustics. London, McGraw-Hill, Chap. 10.
3. *ANON* 1962. Provisional rules for the use of rubber bearings in highway bridges. Ministry of Transport Memo No. 802, H.M.S.O., 1962.
4. WALLER, R.A. 1966. Building on springs. *Nature*, 1966, **211,** No. 5051, pp 794–796.
5. GROOTENHUIS, P. 1967. The attenuation of noise and ground vibrations from railways. *Journal of Environmental Sciences*, 1967, April, pp 14–19.
6. REED, A.J. 1966. Sound and vibration insulation of large structures. *Rubber in Engineering Conference*, Natural Rubber Producer's Research Association, London, 1966, Paper No. 15.
7. MOSS, P. 1983. Isolating appartments and offices from ground-borne vibrations. *Noise and Vibration Control Worldwide*, September 1983.
8. CLAYDEN, K.J.G. 1966. Noise and vibration attenuation in the Barbican Scheme. *Rubber in Engineering Conference*, Natural Rubber Producer's Research Association, London, 1966, Paper No. 16.
9. MATHERS, C.D. *and* WALKER, R. 1985. Some practical aspects of vibration isolation. *Proc. I.O.A.*, **7,** 3, 1985. Institute of Acoustics, Edinburgh.
10. SMITH, T.J.B., WALKER, R. *and* MATHERS, C.D. 1986. Studies on a vibration-isolated radio studio mock-up. *12th International Congress on Acoustics,* Paper **D1–5,** Toronto 1986.
11. BRAMER, T.P.C., COLE, G.J., COWELL, J.R., FRY, A.T., GRUNDY, N.A., SMITH, T.J.B., WEBB, J.D. *and* WINTERBOTTOM, D.R. 1977. Basic vibration control. Sudbury (Suffolk, England), Sound Research Laboratories Ltd., 1977.
12. HANSON, O.B. *and* MORRIS, R.M. 1930. The design and construction of broadcast studios. *Proc. Inst. Radio Engrs.*, 1931, **19,** 1, pp 17–34.
13. BERANEK, L.L. 1958. The transmission and radiation of acoustic waves by structures. London, *The Forty-fifth Thomas Hawksley Lecture*, Inst. Mech. Eng., 1959.
14. MEARES, D.J. 1980. Revised background noise criteria for broadcasting studios. BBC Research Department Report No. 1980/8.
15. REIHER, H. *and* MEISTER, F.J. 1931. The sensitivity of human beings to vibration. Forsch. Geb. Ingwes, **11,** 2.

16. *ANON* 1983. Vibrations: building and human response. *Building Research Establishment Digest 278*, 1983, H.M.S.O.

17. RANDALL, K.E., MEARES, D.J. *and* ROSE, K.A. 1986. Sound insulation of partitions in Broadcasting Studio Centres: field measurement data. BBC Engineering Publication, October 1986.

10. ACKNOWLEDGEMENTS

The authors gratefully acknowledge the contributions made by the Acoustic Consultants, Tim Smith Acoustics and Arup Acoustics Ltd., the Architectural Consultants, Foster Associates, the Structural Consultants, Ove Arup Partnership and the Contractors, Bovis Construction Limited. Also those of the following members of BBC staff. The Langham Project Control Group: A.E. Berry, G.W. Crowe, A.D. James. Acoustics Committee: D.J. Meares and D.G.M. Stripp. Architectural and Civil Engineering Department: K.A. Rose, P.L. Jeffree and F.R. de Zilwa. Research Department: K.J. Wright, M.J. Bright, T.J. Borer and M.J. Knee.

APPENDICES

APPENDIX 1

Outline of Computer Program for the Derivation of Network Responses by Inspection

Consider a generalised ladder network of the form shown in Fig. A.1. At any section, for example that at 'B', the input impedance of the portion of the network on the right-hand side, Z_n, can be written down by inspection in terms of the input impedance of the next segment, Z_{n+1}, together with the intervening impedances, in this case X_7. These impedances may be in series or in parallel, depending on the configuration at the section, for example, Z_{n-1} is given by the parallel impedance of Z_n and X_6. By working from the output end of the network back towards the input an array of input impedances can be derived, one for each section, the derivation being written by inspection at each stage.

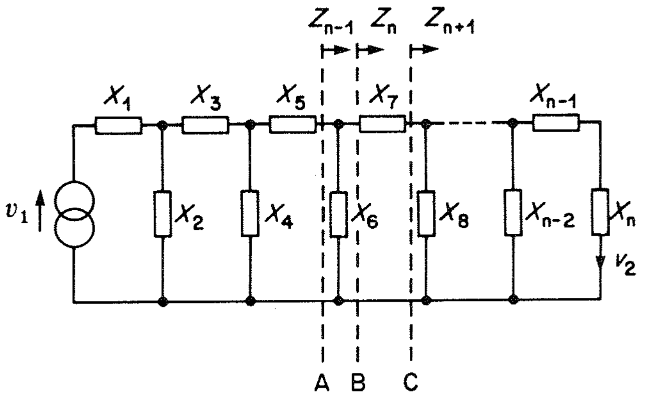

Fig. A.1 – Generalised network.

When all of the impedances have been derived the attenuation factor at each node can be derived as the ratio of the two impedances, for example at 'A' the current divides between the two parallel paths and the proportion Z_{n-1}/Z_n of the input current at that stage goes on to the next stage. The overall current transfer function is simply the product of all of these division ratios.

In a computer language which does not support complex arithmetic, procedures for addition, multiplication, division and the parallel combination of impedances can be prepared in a standardised, 'blank' program. It is then simply necessary to add the procedure for calculation, in terms of these other procedures, written line-by-line by inspection of the circuit diagram.

In breaking down the analysis to single steps, combining never more than two impedances or ratios, the method avoids any requirement for algebraic manipulation. At the same time, no approximations are made so that the result is accurate, within the limitations of the computer resolution. The expression representing the network may involve many steps and perhaps inefficient in the use of processing time but it can be written almost as fast as the keyboard can be operated.

APPENDIX 2

Component Values for the Idealised Equivalent Circuits

The prototype circuit on which all of the calculated responses were based is that representing a drama studio on a set of vibration isolating mountings. The mass of the studio has been assumed to be 450 tonnes. The compliance is defined as the deflection per unit force so, for a deflection of 25 mm, the compliance is $25 \times 10^{-3}/(450 \times 10^3 \times 9.81)$. This is equal to 5.7×10^{-9} m.s^2. The fundamental resonant frequency, f_r, is given by

$$f_r = 1/(2\pi(\text{mass. compliance})^{1/2})$$

giving a value of 3.15 Hz for the prototype system.

The value of the resistance representing the loss of the circuit is given by 1/'Q' times the impedance of the compliance (or the mass) at the resonant frequency. For a 'Q'-factor of 20 for the prototype circuit, this gives a value for the resistance of 444,000 Ohms. In reality, the losses of such systems are not usually represented accurately by a single fixed resistance at all frequencies. However, the difference in the overall character of the response caused by this simplification is not great for the present purposes.

In Figs. 4 and 5 the compliance was split into three equal parts, each equal to one third of the total compliance. The loss components were treated in the same way.

The response of the circuit of Fig. 6 has not been given explicitly but a representative value for the additional compliance, assuming that the case is that of a typical 'noise-stop pad' is 4.6×10^{-10} m.s^2. The additional mass will be approximately that of the vibration isolation mountings and any extra mass added to improve the performance of the 'noise-stop pad'. For the case being considered it is unlikely to exceed 20 tonnes. The additional loss component can be calculated in the same way as above.

The component values for the circuit of Fig. 8 were derived by dividing the load mass by ten and taking a value for the modal compliances to give a lowest modal frequency of about 20 Hz. The value used for C_s was 1×10^{-10} m.s^2.

APPENDIX 3

Types of Mount and Description

Fig. A.3 illustrates the various mount types.

1) Airmounts

A rubber, tyre-like construction, relying on compressed air for the supporting force. They have the advantage of a low and variable (depending on the air pressure) resonant frequency and light, well-damped construction relatively free from unwanted modes. However, it was difficult, even on the experimental structure, to avoid leaks. Over a period of time such as that demanded by a 'permanent' building structure a system of continuous measuring and re-inflating would be necessary. The samples tested had a maximum load capability, at the design height, of about 6 tonnes each. They were, however, used at a lower load and pressure to give a resonant frequency of about 2.5 Hz. Two arrangements of these mounts were tested, with four and with eight supports respectively.

2) 'Large' steel springs

Two sets, one of four and one of eight steel helical springs with a maximum loading of 4 tonnes and 2 tonnes respectively were tested. Both sets were loaded to approximately 0.7 times their maximum deflections, giving resonant frequencies of about 3 Hz. The problems of resonant modes in these types of mounts have been covered in the main text.

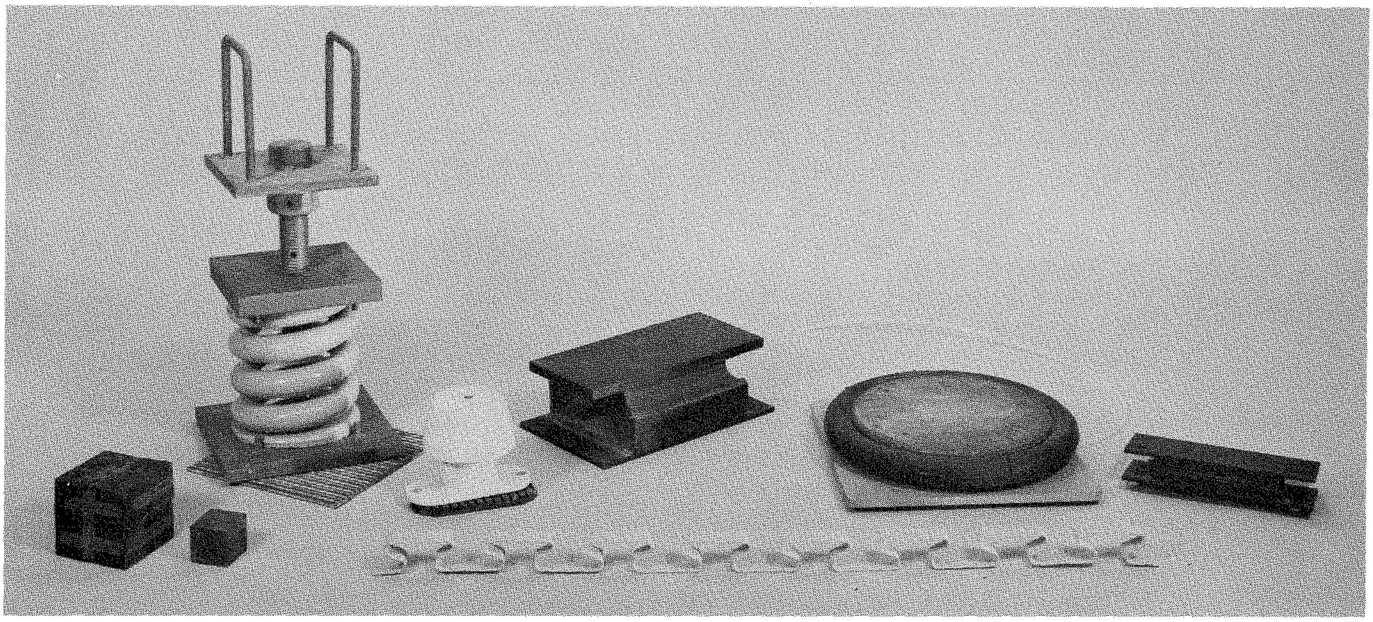

Fig. A.3 – Selection of mount types tested.

3) Rubber in compression

Two arrangements of mounts using rubber in compression were tested, one using four mounts and one using eight. Both sets of mounts were loaded to near their maxima but their design was such that the static deflection was only about 2 mm, giving resonant frequencies of about 11 Hz. To be directly comparable with the other types they should have been designed with much larger working deflections. The problem of resonant modes in these types of mounts is reduced by their simple construction and the small amount of comparatively lightweight material, together with the short conduction path length. The material is also inherently damped to some extent. As described in Section 3.3, when measured without any modal behaviour in the supported mass, these samples showed close to the theoretical isolation performance, that is, no inherent modal behaviour, up to at least 400 Hz.

4) Rubber in shear

Two arrangements of mounts using rubber in shear were tested, using the same mounts for both. The mounts were designed to have a maximum load of about 3 tonnes and a static deflection of about 25 mm. When used in a set of four to support the Test Slab they were nearly fully loaded. However, the load/deflection characteristic of this type of mount is significantly non-linear so the fundamental resonant frequency was not as low as the static deflection would indicate. A value of about 4.5 Hz was measured. The second arrangement tested, using eight of the same mounts, was underloaded and gave a resonant frequency of about 6 Hz. Like those using rubber in compression, the construction and material of these mounts should make them relatively free from spurious modes. However, the samples tested had large steel plates attached which were impossible to restrain from vibration and it was not possible to confirm this. In a practical construction, the steel plates would be constrained by the material of the structure and would probably not vibrate independently.

5) 'Small' steel springs

A system of a larger number of smaller steel springs, using a total of 27 units was also tested. These springs were proprietary units with a maximum design loading of about 0.5 tonnes and a maximum static deflection of 20 mm. The design included some damping of the fundamental resonance but it was not clear whether the damping mechanism had any effect on the spring modes. It appeared not to have. The fundamental resonant frequency was, as expected from the static deflection, about 4 Hz.

6) Glass-fibre cubes

A component of a proprietary floor structure, primarily intended to improve the airborne sound insulation of a floor at the same time as providing a load-bearing capability. Made of small glass-fibre cubes, 50 x 50 x 50 mm, which can also serve as vibration isolation mountings. Two arrangements of these were tested, one with 108 cubes and the second using 28 cubes of double the dimensions in each direction (that is, eight times the volume), evenly spaced on rectangular matrixes. These cubes, being very light, should exhibit no low-frequency modes, although for experimental reasons it was not possible to confirm this above about 150 Hz. The maximum static deflections are, however, small and so the resonant frequencies were about 14 Hz and 10 Hz respectively.

7) Steel strip spring

A proprietary type of vibration isolator consisting of corrugated stainless steel strip. The type tested was about 30 mm wide with overlapping corrugations to make the height about 50 mm. The maximum rated static deflection was only 6 mm so isolation at low frequencies could not be obtained. The structure was, however, very lightweight, so that unwanted resonances only occurred at high frequencies.

Printed by BBC RESEARCH DEPARTMENT, Kingswood Warren, Tadworth, Surrey, KT20 6NP